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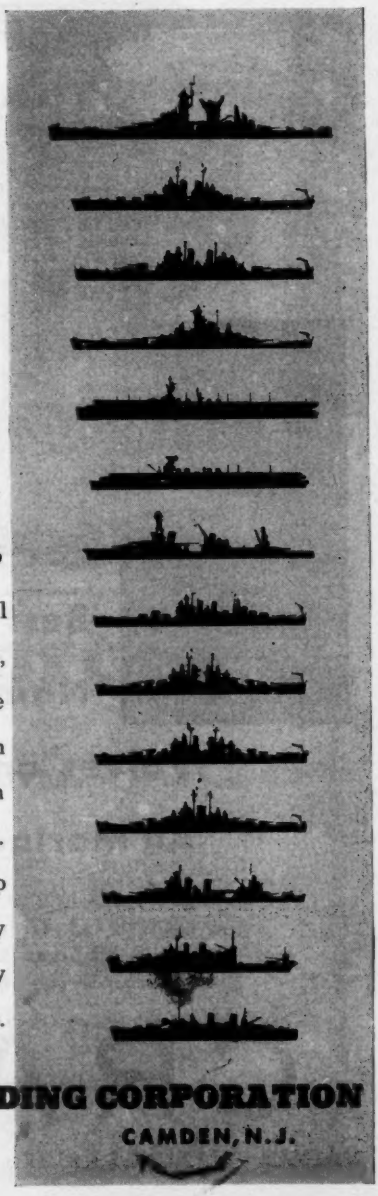

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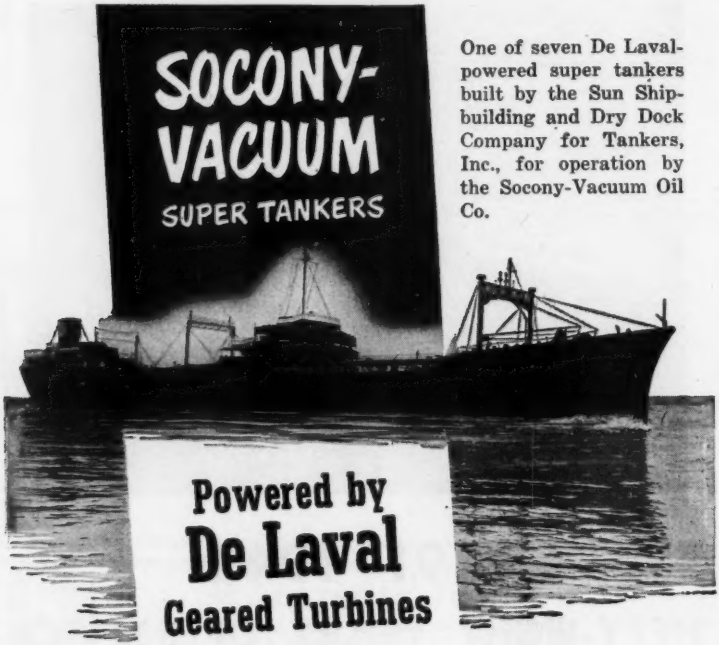

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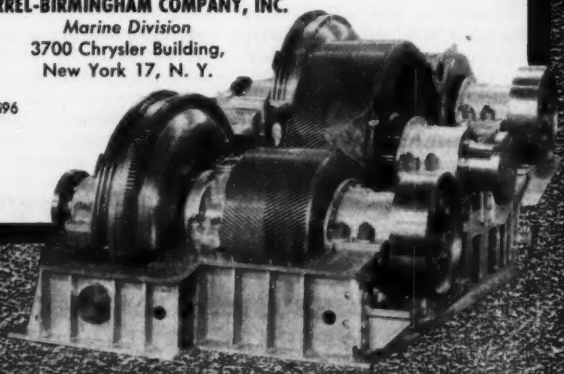

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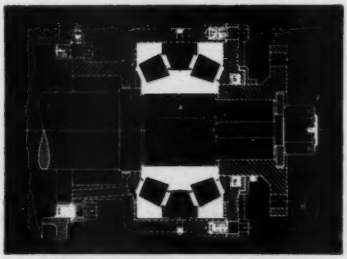
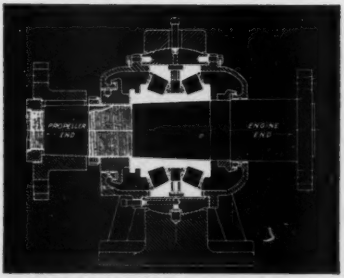
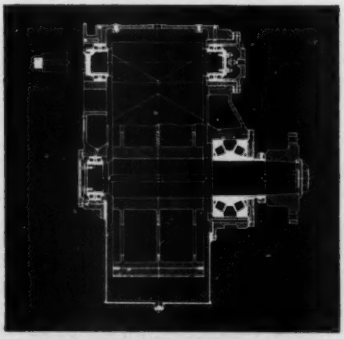


Timken propeller thrust bearing mounted on an extension of the engine crankshaft and bolted to the engine case.



Timken bearing separate thrust block for use between engine and propeller shaft.



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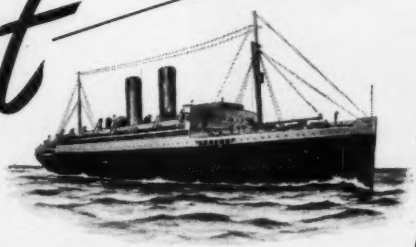

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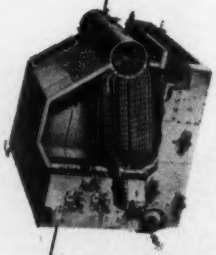

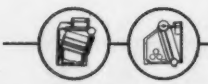

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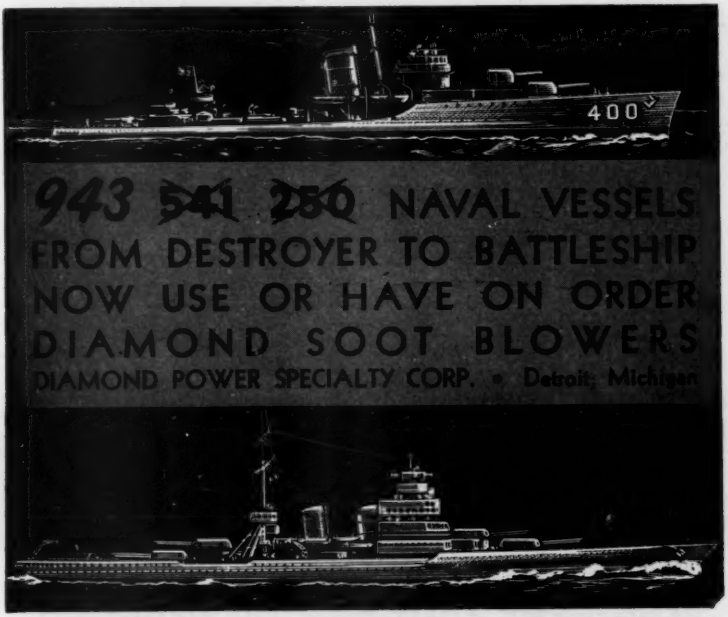
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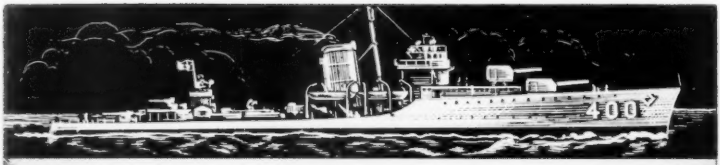

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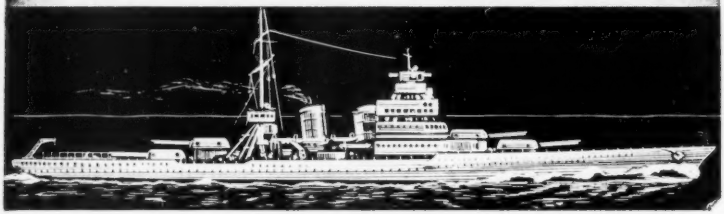

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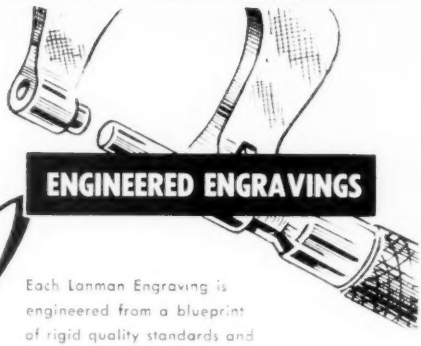

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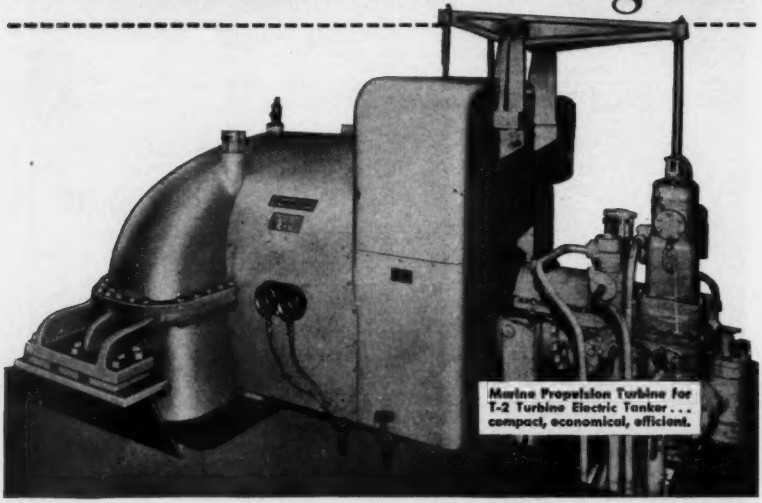

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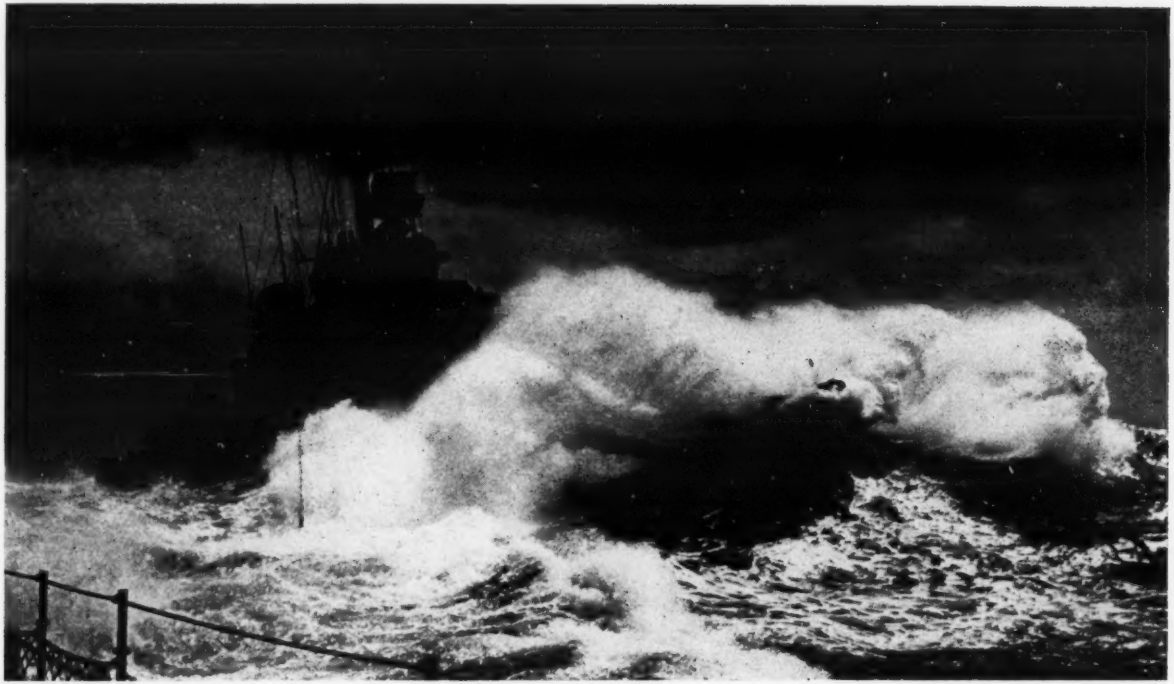

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Destroyer Noses Under a Heavy Sea—One of the unpredictable elements which must be considered in ship design and construction.

Destroyer Noses Under a Heavy Sea—One of the unpredictable elements which must be considered in ship design and construction.

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JOURNAL

OF

THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

VOL. 62. MAY 1950. No. 2

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SECRETARY'S NOTES

Council Change

Captain Homer Ambrose, U.S.N., has been detached from duty in Washington. In accordance with the by-laws he thus becomes ineligible to complete his term as a Regular Naval Member of the Council. The Council has appointed Rear Admiral T. C. Lonnquest, U.S.N., to fill the vacancy and Admiral Lonnquest has accepted.

Admiral Lonnquest is Deputy and Assistant Chief of the Bureau of Aeronautics. Previously he has served as a member of the Council. His new term will expire on 31 December 1950.

Life Membership

The Council has adopted rules and rates for the Life Membership which was voted into the by-laws at the last election.

The rates were selected at a level which offers no particular financial advantage to members who desire the convenience of Life Membership. Since the purpose of the amendment was convenience and not to strengthen the treasury of the Society, the Council deemed this to be proper. The rates will be found in Association Notes on page 548.

Local Chapters

The rules for the establishment of Local Chapters are published in Association Notes on page 549. These rules were prepared by a sub-committee appointed by the President and were adopted by the Council on 13 April 1950.

Applications for recognition of Local Chapters should be addressed to the Secretary-Treasurer. It is suggested that a copy of proposed local by-laws, and list of charter members accompany the letter application to expedite action.

Annual Banquet

The 1950 annual banquet was held at the Hotel Statler, Washington, D. C., on Friday, 28 April 1950. This meeting was marked with the same success which has characterized each of the banquets since the war. Contrary to some expectations, attendance was the highest ever, exceeding the record established last year. The limiting capacity of the Statler which is the largest in Washington has now been reached.

In order that members who were unable to attend, may have an opportunity to read the valuable address which was presented by Mr. David Sarnoff, Chairman of the Board of the Radio Corporation of America, it has been printed and is mailed with this issue of the Journal.

The Journal

The last few issues of the Journal have carried a diverse group of articles. Comment and opinions from readers on the nature of the selection will be very helpful to the Editor in pre-

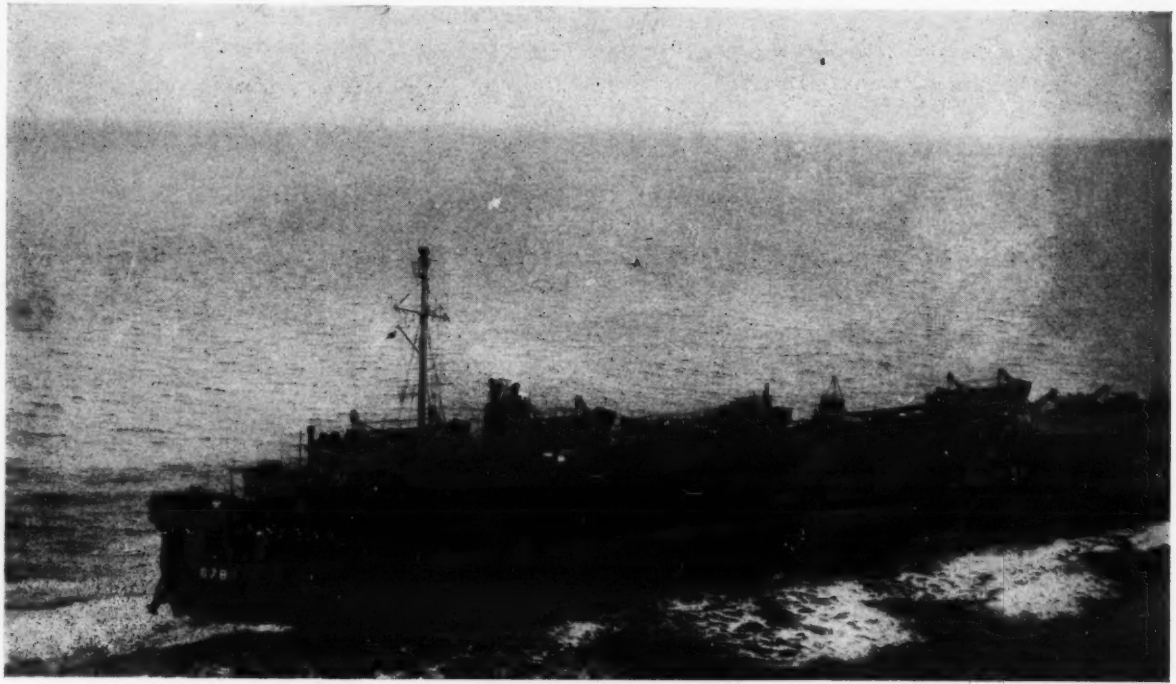
paring future issues. This assistance is second in importance, only to the receipt of original manuscripts which are always welcome.

The February and May 1950 issues of the JOURNAL appear in the format which has now been adopted by the Council as standard. No change in format is intended in the future unless sufficient adverse comment is received to cause the Council to reconsider the matter.

As a matter of interest, the Editor has reviewed Volume I of the JOURNAL. This was the issue of 1889. The four issues contained 23 original articles by authors, many of whom were or became distinguished Naval Engineers. The total content of the first volume on a word-count basis was about 10 per cent less than that of each of the last four single issues. We have thus established a capacity for fostering original naval engineering thinking at about five times the original capacity. This is intended as an invitation to today's and tomorrow's outstanding naval engineers to offer the fruits of their originality to the Society.

MEMBERSHIP LIST

The new membership list has been mailed. In case errors appear in the listing of any member who filled out and returned the data form, the editorial staff will acknowledge full responsibility.



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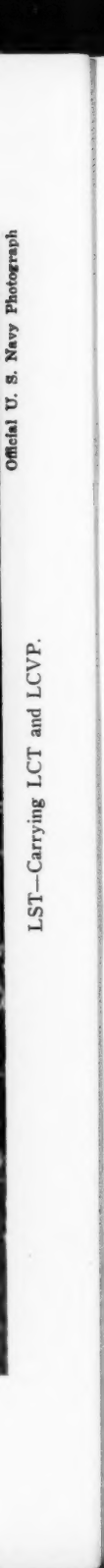
LST—Carrying LCT and LCVP.



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PART ONE

ORIGINAL ARTICLES

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part One are the original works of the authors.

They are offered in furtherance of the Society's raison d'etre: to further the advancement of naval engineering. Discussion of any article is invited for publication in a subsequent issue.

Permission to reprint is freely granted on condition that *full acknowledgment is given* to the author and to the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

All articles which have been written by naval personnel and are published herein carry the personal view of the authors and, unless specifically so stated, do not express the official views of the Navy Department.

Official U. S. Navy Photograph

Port bow aerial view of the aircraft carrier U.S.S. *Lexington.*

OPERATIONAL STABILITY OF FREE PISTON GASIFIERS

R. HUBER

THE AUTHOR

is Technical Director of Bureau Technique Pescara in Paris. He is a native of Switzerland and graduated from the Federal Institute of Technology in Zurich some twenty-five years ago, a favorite pupil of the renowned Stodola. That spring a French inventor came to Stodola looking for a "bright young man" to engineer some of his revolutionary ideas on a light propelling plant for a helicopter. It was fortunate that R. Pescara, the dreamer and glib-tongued persuader, chose Huber the engineer, who was young and capable and devoid of any inhibitions on the constructional details to be used in developing the new free piston machinery. Under his supervision over twenty-five different sizes and types of free piston machinery were designed and seventeen were built and tested. The war added tribulations to the new enterprise located as it was in Paris but it also stiffened the determination of these pioneers. At war's end a new start was made and this time the world appears to be ready to recognize the merits of the free piston gasifier. The JOURNAL deems it a privilege to offer its readers R. Huber's original analysis on operational stability of free piston gasifiers—a subject on which many honest and conscientious engineers have apprehensions.

The free piston gasifiers or gas generators feeding a gas turbine have found industrial application in France.

To this new type of heat engine, the Diesel engine furnishes the advantage inherent to it, i.e. combustion at high efficiency, while the gas turbine on the other hand develops mechanical energy under optimum conditions of exploitation of structural materials, thus with a very high power factor per unit weight.

The cycle is characterized by an initial expansion of the combustion gases in a reciprocating engine followed by expansion in the turbine which produces the useful energy. This cycle has been achieved in practice by using a highly supercharged internal combustion cylinder in which the supercharging was produced by a compressor driven by the engine itself.

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Port bow aerial view of the aircraft carrier U.S.S. *Lexington*.

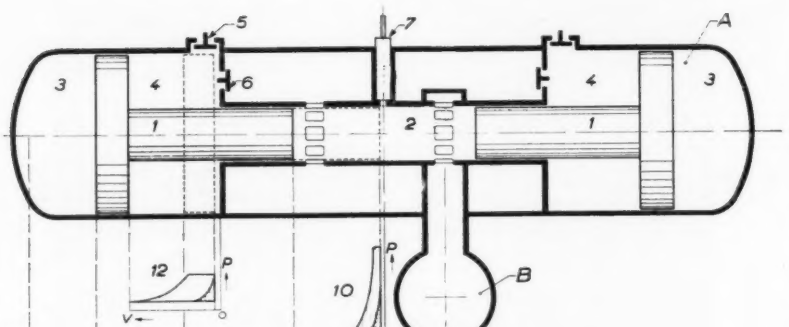

Figure I—Diagrammatic Sketch of a Free Piston Gasifier and Gas Turbine Including Pressure Volume Diagrams of the Power, Compressor and Cushion Cylinders.

A—Gasifier
B—Receiver
C—Gas turbine

1—piston
2—power cylinder
3—cushion or bounce cylinder
4—compressor cylinder
5—intake valves
6—discharge valves
7—fuel injector
10—p-V diagram of power cylinder
11—p-V diagram of cushion cylinder
12—p-V diagram of compressor

Free piston gasifiers lend themselves particularly well to the attainment of this cycle. Absence of a mechanical limit to the stroke of the pistons permits adapting the position of the inner dead center to the degree of supercharging of the cylinders. Furthermore, the thermal effect of the gases on the cylinder walls is reduced, owing to the high acceleration of the pistons during combustion which reduces the time during which the walls are subjected to gases at high pressures and temperatures.

Among the problems to be solved in the course of development of these machines one of the most difficult was concerned with the absolutely uniform and constant operation of a piston not attached to an assembly consisting of connecting rod and crankshaft. At first sight it may appear that elimination of the classical connecting rod and crankshaft linkage, used at present in all internal combustion engines would necessarily culminate in a sensitively balanced operation, and that minor disturbances or misadjustments would cause the engine to operate irregularly and even cause it to stall, i.e., would "kill" the engine. However, many gasifiers of this type were built up to the present and have run for thousands of operating hours without a single incident attributable to the absence of wrist pins, con-rods and crankshafts.

STRUCTURE AND OPERATION OF THE GASIFIER

The gasifiers used for the tests to be discussed herein have been described elsewhere in great detail. Therefore, it will be sufficient in the following to consider only briefly the fundamental concepts involved and the operation of the unit, while measures taken to insure stable operation will be studied in greater detail.

The gasifier arrangement finally adopted after testing various other types is shown in Figure 1.

The gasifier A contains two opposed step pistons (1) of equal and symmetric stroke. The power cylinder (2) is in the center. It operates as a two-stroke Diesel engine supercharged to a pressure of several atmospheres. The two single-acting compressor cylinders (4) are located on both ends of the central housing. The end spaces (3) constitute the cushion or bounce chambers which store the energy for the return stroke. The compressed air is used for scavenging and charging of the power cylinder of the unit. Fresh air is aspirated through valves (5) and is discharged through the valves (6) into the space surrounding the power cylinder. The fuel is injected by several injectors (7), mounted in the center of the combustion chamber.

The hot combustion gases, mixed with the excess scavenging air, exhaust into receiver B and then flow through the gas turbine C which alone produces the useful power, i.e., is the prime mover of the unit.

During the outward stroke of the pistons all of the energy developed by combustion in the power cylinder, augmented by the re-expansion of air in the clearance space of the compressor cylinders, is stored in the cushion chambers (3). The work developed during this stroke is represented by the vertically cross-hatched areas in the pressure volume diagrams (10) and (12) for the power and compressor cylinders. This energy is subsequently recovered, except for friction losses, as shown in the p-V diagram (11) of the cushion or bounce cylinders.

During the inward stroke the air compressed in the cushion chambers drives the pistons back toward the center with a power sufficient (a) to compress and discharge the air in the compressor cylinders and (b) to compress the air used for combustion in the power cylinder. The work of this stroke is illustrated schematically by the areas cross-hatched horizontally in diagrams (10), (11), and (12) of Figure 1.

The air pressure in the cushion cylinders for different load conditions is regulated by a stabilizer designed so that the mean pressure in the cushion is independent of the stroke and varies linearly with the discharge pressure of the compressor.

The distance of the outer dead-center from the central plane through the injectors increases with increasing output. On the other hand, the work absorbed by the compressor grows also with distance to the outer dead-center, since the quantity of air aspirated is increased. For differing positions of the outer dead-center, the energy accumulated in the bounce chambers and the energy required by the compressors vary in the same sense and approximately by the same value, so that the energy required for compression of air for combustion is practically constant. Therefore, for a given output gas pressure, nearly the same end pressure in the power cylinder is obtained for all positions of the outer dead-center.

The two pistons are connected by a mechanical linkage which equalizes the slight, inevitable inequalities due to friction, wear, and intake air pressure. However, this synchronizing gear does not limit the stroke of these free pistons either inward or outward.

Since this synchronizing gear cannot prevent the pistons either from colliding with each other or from impact with the

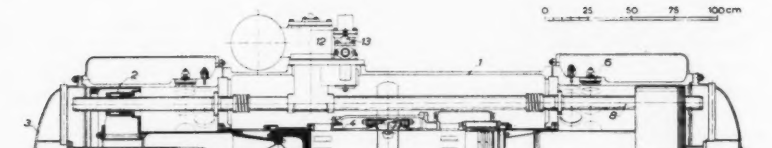

Figure 2.—Vertical Section Free Piston Gasifier, Model GS-34.

1—housing
2—compressor cylinder
3—cylinder head
4—power cylinder
5—scavenging air receiver
6—intake valves
7—discharge valves
8—piston guide
9—cooling oil inlet
10—cooling oil discharge
11—stuffing box
12—starter
13—stabilizer
14—exhaust duct

cylinder heads at the end of the outward stroke, it is obvious that absolutely infallible means must be provided to prevent such accidents which would lead to destruction of the machine. The gasifier may be equipped with a regulator to maintain a desired stroke for instance. However such a regulator would have to respond at very high speed, as the stroke may vary in length from one cycle to the next, in less than 1/10 second. It is to be feared, that the response of such regulators would be too slow.

It seemed more desirable to design a gasifier without reliance on stroke regulators or the like and to assure absolute reliability of operation by judicious dimensioning of the parts and by proper balancing of power compressor and bounce cylinder sizes.

The gasifiers considered in the following study were all designed according to the foregoing principle. Figure 2 shows a longitudinal section through one of these gasifiers, Model GS-34, with principal characteristics and dimension as follows:

Power cylinder bore	340 mm	(13.386 inches)
Compressor cylinder bore	900 mm	(35.433 inches)
Maximum stroke possible	550 mm	(21.654 inches)
Useful stroke at full load	443 mm	(17.441 inches)
Number of cycles per minute	613	
Gas pressure at full load	3.5 kg/cm²	49.70 psig
Gas temperature	507°C	(944°F)
Thermal efficiency of gas	41.5%*	
Gas horsepower	1350 H. P.*	

* Based on adiabetic expansion.

REGULATION

Variation of the horsepower of the turbine fed by one or more gasifiers is obtained by simultaneous variation of pressure and output of the gases produced by the gasifiers. Thus all efficiency losses resulting from turbulence ahead of the turbine are avoided.

As shown in Figure 4, the turbine speed governor (1) acts directly on the fuel injection pump (2) of the gasifier movable stop (3) between the two cam faces (4) is subject to gas pressure acting on the piston (5) and fixes the limits of the stroke of the fuel injection pump and thus adapts the maximum and minimum limits of possible fuel charge to the range of operation as determined by the gas pressure to the turbine.

The frequency or number of piston cycles varies but slightly with the load. To adapt the gas output to changes in turbine load, the position of the outer dead-center must therefore be shifted—it must be moved away from the center as the power increases. In the gasifier being considered, the outer dead-center may occupy any position ranging from 360 to 550 mm (14.17 to 21.65 inches) from the center of the machine. However such a great variation is not necessary for adapting the output of the gasifier to the consumption of the turbine. In actual practice, the dead-center positions fall within a range between 380 to 480 mm (14.96 to 18.90 inches) from the center of the machine.

Figure 3.—Outside View of Free Piston Gasifier, Model GS-34.

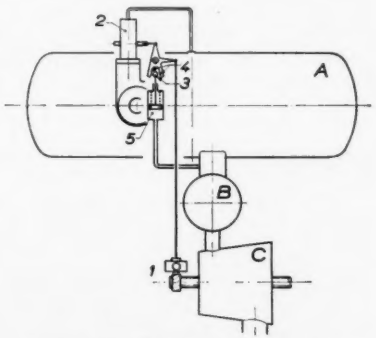

Figure 4.—Schematic Arrangement of Controls.

A—gasifier
B—receiver
C—turbine

1—speed governor
2—fuel injection pump
3—movable stop
4—minimum and maximum output cam
5—movable stop control piston

PISTON STROKE RECORDS

The records of successive piston strokes offer a very accurate means for analysis and study of the operation of the gasifier without parallel or equivalent in crank-connected machinery. These records show the position of the dead-centers, both inner and outer, very accurately with respect to the center of the machine from which may be determined the compression pressure within the power cylinder, the instantaneous load carried, as well as the regulation of successive cycles. The stroke indicator used on these gasifiers includes a stylus connected rigidly to one of the pistons and records on a rotating drum the

actual length of stroke—see Figure 5.

A stroke diagram with constant fuel injection is reproduced in Figure 6. It will be noted from this diagram that the regularity of stroke is very high. The maximum deviations of the outer dead-center with respect to the mean position do not exceed 1.4 mm (± 0.05512 inch). For these positions of the outer dead-center, the effective pressure in the bounce cylinder is equal to 4.6 kg/cm² (65.32 psig). The energy variation due to these irregularities is 41 kgm (296.55 ft lbs) and, since the indicated output equals 3430 kgm (2480.9 ft lbs), these fluctuations will be of the order of ± 1.2%. In reality the power output varies even less, since the displacements of the outer dead-center are not caused solely by the small differences of gasifier output from one stroke to the next, but include also the variations of the compressor work and friction losses.

Figure 5.—Piston Stroke Indicator.

Such an observation would be difficult, if not impossible, to make on a standard Diesel engine, because an indicator card for a single power cycle cannot be taken

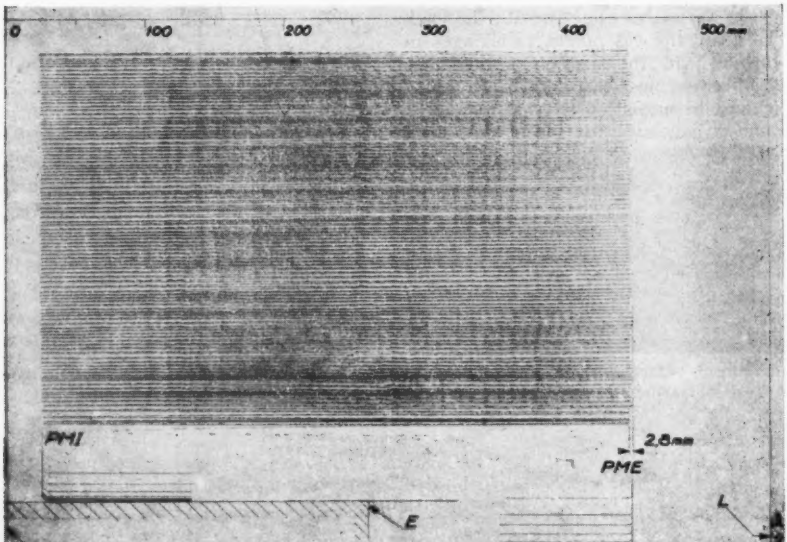

Figure 6.—Stroke Diagram with Constant Fuel Injection.
P.M.I. —inner dead-center
P.M.E. —outer dead-center
C —center of the machine
L —outer stroke limit
E —edge of exhaust ports

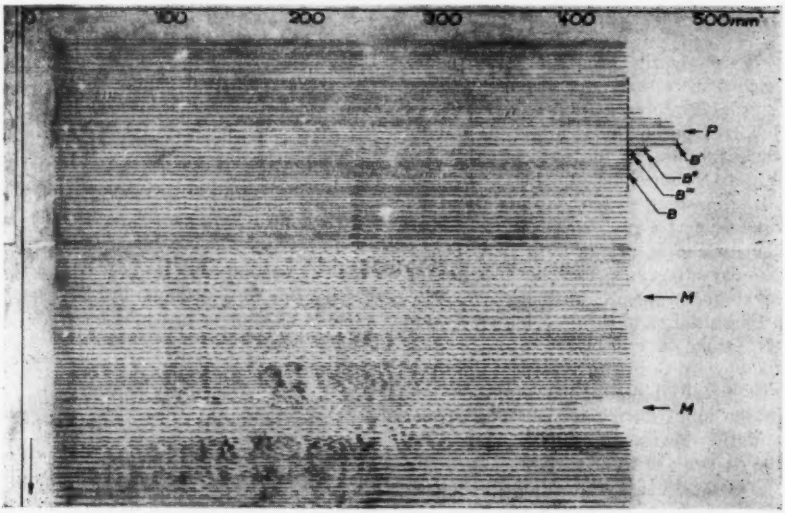

Figure 7.—Stroke Diagram with Momentary Increase and Decrease of the Quantity of Fuel Injected.

P—increase in fuel injected
M—decrease in fuel injected
B—position of dead-center, standard operation
B′, B″, B‴—position of dead-center determined graphically on Figure 9

with sufficient accuracy to avoid deviations of about 1% between two diagrams. Evaluation of numerous stroke diagrams indicates that the maximum fluctuations of the motor and compressor work are less than 1 per cent. These diagrams reveal, furthermore, the very great regularity of operation of the fuel injection equipment as well as the constancy of friction.

Stroke records constitute, moreover, a very practical means of tracing the changes resulting from momentary disturbances, notably injection troubles. For test purposes, such disturbances are reproduced by varying the amount of fuel injected for a period of several cycles. The stroke diagram shown in Figure 7 shows the records taken under such conditions; P denotes the case for an increase of the stroke, whereas M shows a decrease of the latter. The displacement of the outer dead-center was in both cases approximately 40 mm (1.5748″). Moreover, at the instant of cessation of this disturbance, the original, normal dead-center is restored within a few strokes without the least instability effect, causing, for example, a fluctuation of the outer dead-center about some mean position. These diagrams show, therefore, that there is a very pronounced tendency on the part of the gasifier to readjust itself to standard operation after each interruption of normal running. In accordance with standard terminology, this tendency will be termed *Operational Stability* of the gasifier. The degree of stability of a gasifier depends on the arrangement and on the dimensions of the power compressor and bounce cylinders.

ENERGY DIAGRAM OF A GASIFIER

To study the behavior of a gasifier with respect to its operational stability, a graphic representation of the energies involved may be used to permit tracing the evolution of the successive cycles. Figure 8 shows such an energy diagram for a gasifier, based on Model GS-34 operating at a gas discharge pressure to the turbine of 2.5 kg/cm² (35.5 psig). Similar diagrams can be plotted for other operating pressures.

The abscissa of the diagram represents the stroke of one of the pistons, while the upward ordinates represent the energies involved per piston (for one side of the unit) during the outward stroke and the downward ordinates represent the energies of the return stroke.

The total energy including the expansion energy of the air in the clearance space of the compressor during the outward stroke is represented by a family of curves *a*, as a function of the various possible positions of the inner dead-center and for different values of excess combustion air.

This value is characterized by the ratio:

$$\lambda = \frac{\text{Actual weight of air available for combustion}}{\text{Theoretically required weight of air}}$$

Thus $\lambda = 2$ corresponds to a 100% excess of air.

The limits X_1 and X_2 between which the inner dead-center may vary, correspond to final compression pressures of combustion air in the power cylinder of 30 and 100 kg/cm² (426 and 1420 psi). The value of 100 kg/cm² may appear very high, but tests have shown that the absence of a mechanism subject to power cylinder pressures makes operation perfectly possible, even under such conditions. These curves are based on heat losses during combustion determined experimentally as a function of compression pressure and of λ.

The power supplied at constant pressure during the scavenging cycle depends on the length of the stroke, i.e., the remoteness of the outer dead-center from the midpoint. This is apparent from the card of power cylinder shown in Figure 8. To simplify the graph, curves *a* may be made independent of the stroke length by deducting from the sum of the energies the work corresponding to the horizontally cross-hatched portion and by decreasing the energy of the cushion by the same amount, as represented by curves *b* and *c*.

These two curves *b* and *c*, symmetrical with respect to the axis of the abscissas, in reality represent the exact value only for a given inner dead center. However, the slight displacement of this dead-center permits neglecting its effect on the slope of the curve. The curves d_1 and d_2 represent, as function of the inner dead center, the energy absorbed by the power and compressor pistons during the return stroke. Curve d_1 is valid for the position of the outer dead center at 360 mm (14.173 inches) from the midpoint, and d_2 for a distance of 550 mm (21.654 inches) from the center. An intermediate curve *d* between d_1 and d_2 corresponds to a curve *c*, for the bounce energy as a function of the outer dead center.

The points of intersection of d with d_1 and d_2 were obtained by drawing, for the positions of the outer dead center of 360 and 550 mm (14.173 and 21.654 inches), the connecting lines l_1 and l_2. These lines are slanted to consider energy losses by friction which were determined by tests as equal to 740 kgm per meter of stroke. The drawing shows the angle of inclination corresponding to this friction value.

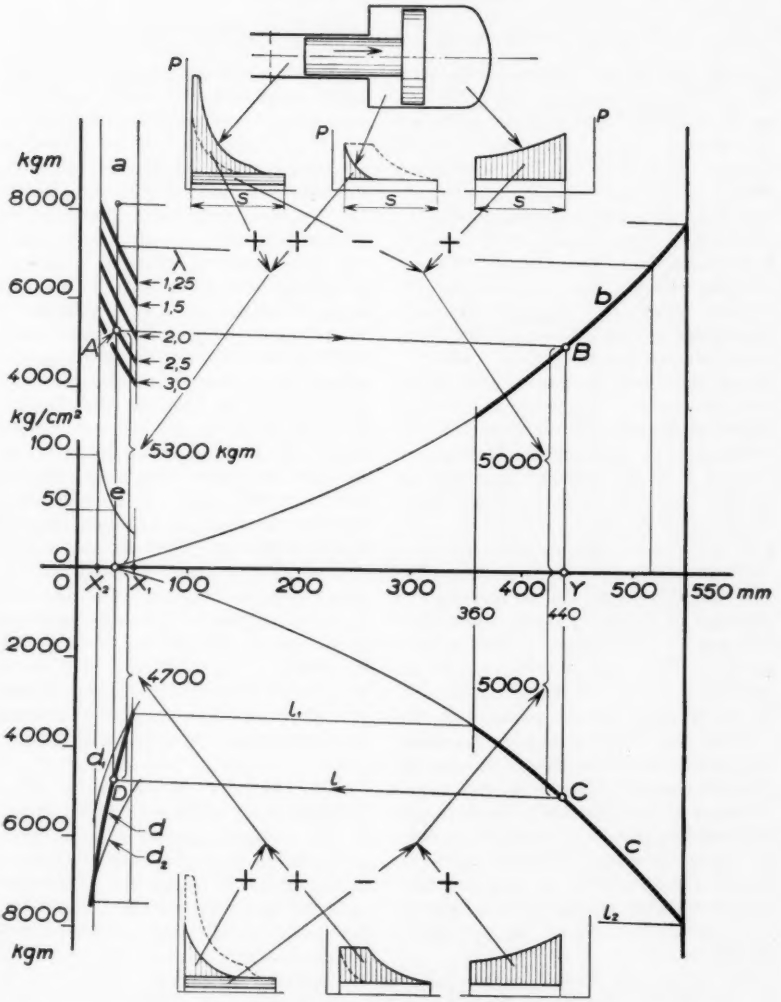

Figure 8.—Energy Diagram of a Gasifier

a —curves of power stroke energy as a function of and the position of the inner dead center
b —curve of power stroke resistance
c —curve of return stroke propulsive energy
d,d_1,d_2—curves of return stroke resistance
e —final compression pressure as function of the position of inner dead center
P —pressure
S —stroke
X_1,X_2—position of inner dead-center for compression pressures of 30 and 100 kg/cm² (426 and 1420 psi)
l,l_1,l_2—connecting lines for 740 kg m per meter of friction stroke
Y —position of outer dead center

The scale of energy refers to a single piston of Model GS-34.

ENERGY DIAGRAM FOR A NORMAL RANGE OF OPERATION

Once the above diagram has been plotted, a cycle can be traced in the opposite direction. For example, by fixing the power cylinder compression at 50 kg/cm² (710 psi), or the inner dead center at 34 mm (1.3386 inch) from the center, we proceed from point (D) on curve *d* toward point (C) on curve *c* by connecting these two points by a slanting straight line, to allow for the friction. From (C) we draw the perpendicular to find (B) and, from there, a new slanted line toward (A) which is found likewise at 34 mm (1.3386 inch) from the center. The value of λ for the motor in this case proves to be 2.5.

This energy diagram reveals in a very concise form the energies involved for normal operation. It also permits determination of energies required to reach the stroke limits mechanically possible, for both power and return strokes. This diagram also shows that to increase the distance of the outer dead center from 440 mm (17.323 inches) to the extreme point of 550 mm (21.654 inches) an additional energy of 68% would have to be developed in the power cylinder.

If we admit that the power developed can never exceed the value indicated by the curve at $\lambda = 1.25$, because for lower excess of air, combustion could no longer be complete, then it is obvious that, even if a greater quantity of fuel were injected, the pistons would still stop well before reaching the mechanical limit of their stroke for lack of air to support combustion.

In reality, the danger of an impact at the end of the stroke is even less grave, since the quantity of fuel injected could never attain the high value corresponding to $\lambda = 1.25$. Furthermore, we have shown in Figure 4 that the quantity of fuel injected per stroke is limited by a mechanical device. Hence, for the operating case shown in the energy diagram, λ could never exceed the values ranging from 1.8 to 3.5.

Another important fact is apparent from this diagram: it is impossible for the pistons to hit each other at the inner dead center even if the cushions were to return the pistons with the maximum energy possible. Should the extreme outer dead center of 550 mm (21.654 inches) be reached accidentally, the compression pressure in the power cylinder will scarcely exceed 100 kg/cm² and therefore the pistons will stop about 40 mm (1.5748 inch) from the center. Furthermore, it is apparent from the very sharp slope of the curve *d* that the position of the inner dead center varies but little, even if the return stroke energy is variable as a result of a shift in the position of the outer dead center. Hence, for all strokes, power cylinder compression pressure is of practically constant value.

The basic conclusion to be drawn from the study of these energy diagrams is the absolute impossibility of impact between the power pistons or between the compressor piston and the bounce cylinder head. Furthermore, this absolute safety of operation is obtained solely by the correct choice of dimensions and by selection of proper capacities for power, compressor and bounce cylinders.

ENERGY DIAGRAM FOR BRIEF DISTURBANCES

Energy diagrams on the other hand may also serve to check the degree of operational stability of a gasifier, this degree being characterized by the number of cycles of operation required for the machine to return to the state of operation prevailing before the brief disturbance occurred, without intervention of the governor as shown on the stroke diagram, Figure 7.

Figure 9 shows the energy curves as a function of the positions of the inner and outer dead centers which appear already in Figure 8. Operation before the disturbance is defined by the lines A-B-C-D of the energy cycle, with $\lambda = 2.5$ and the outer dead center at 440 mm (17.3230 inches) from the center of the machine.

Assuming now that fuel injection be increased by 39% of the quantity injected during a single stroke, then point A shifts to A′ and the stroke lengthens to B′ which is situated at a distance of 477 mm (18.780 inches) from the center. From C′ the pistons are bounced toward the inner dead center with more energy. They stop at D′, located 8 mm (0.315 inch) closer to the center of the machine than previously. During the following stroke this disturbance of normal operation is assumed to cease. Thereupon, the cycle follows the lines A″-B″-C″-D″. The outer dead center has now come closer to center by 14 mm (0.5512 inch) and the displacement of the inner dead center is no more than 3 mm (0.1181 inch). During the second stroke following this momentary increase of fuel injected the pistons are at B‴ and have practically resumed the original state of operation.

The points B′, B″, and B‴, determined graphically with respect to the position of the normal dead center B, are indicated in Figure 7. The degree of stability, as evident from the diagram in Figure 9, is determined by the slope

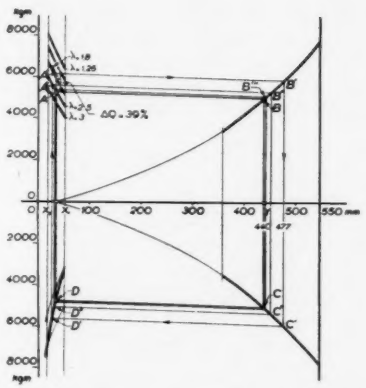

Figure 9.—Energy Diagram of a Gasifier

A-B-C-D	—cycle before disturbance
A′-B′-C′-D′	—cycle during disturbance
A″-B″-C″-D″	—first cycle after disturbance
B‴	—position of outer dead center for second cycle after disturbance
ΔQ	—momentary increase of the quantity of fuel injected in %

of the curve *d* with respect to the abscissa, as compared to the slope of the curves *a*. It can be seen that if, as a result of an improper dimensioning of the various cylinder capacities involved, the curves *d* were of lesser slope than curves *a*, unstable operation would result and the slightest maladjustment would cause the gasifier to stall.

ENERGY DIAGRAM WITH CONSTANT MEAN PRESSURE IN BOUNCE CYLINDER

Up to the present, we have assumed that during the various cycles examined the pressure diagrams and energy of the bounce cylinders remained constant, or that the air mass trapped in the bounce cylinders remained invariable. This assumption is true only as far as very brief disturbances of normal operation are concerned. However, if the stroke variations persist through several successive cycles, the governor which regulates the air pressure in the bounce cylinders has time to respond and to change the air-mass to suit new operating conditions. This stabilizer is designed, as mentioned previously, to maintain in the bounce cylinders a mean constant pressure independent of the

length of the stroke for any given compression pressure. If an increased gas output is needed and the stroke lengthens for several cycles and the mean pressure rises, the stabilizer will then permit a small quantity of air to escape from the bounce cylinder to restore the original mean pressure. The energy accumulated in the bounce cylinder at constant mean pressure is represented on the energy diagram, Figure 10, by curves b′ and c′ which replace curves b and c such as would have been obtained for the case of a constant quantity of air in the bounce cylinder.

For the curve c′, representing the energy of the bounce cylinder for the return stroke, there corresponds a curve d′ for a given value of the friction. It will be noted that in this case the stroke variations affect the compression stroke pressures even less than in the diagram, Figure 8. Thus, for all positions of the outer dead centers between 360 and 550 mm (14.173 and 21.654 inches), the compression pressure in the power cylinder will vary only between 42 and 55 kg/cm² (596.4 and 781.0 psi).

On Figure 8 it has been shown that in no case could the pistons reach their possible stroke limit in outward direction. On the other hand, Figure 10 showing the energy diagram for constant mean bounce cylinder pressure could lead us to believe that an impact between the piston and cylinder head may occur, since for $\lambda = 1.4$ the pistons could attain the limit of possible stroke under certain conditions. In reality, this hazard can easily be avoided by including a safety device, similar to a turbine overspeed governor which would stop the gasifier whenever an arbitrarily set stroke length were exceeded.

Such a safety device would not be an absolutely certain guarantee if it had to respond between two successive strokes. However, for operation at constant mean bounce cylinder pressure as represented in Figure 10, the time-base requisite to vary the air-mass in the

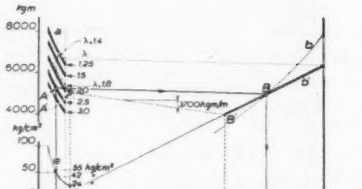

Figure 10.—Energy Diagram for Constant Mean Pressure in Bounce Cylinder

A-B-C-D	—cycle at normal friction
A′-B′-C′-D′	—cycle at maximum friction
b, c	—energy of constant air mass in bounce cylinder
b′, c′	—energy with constant mean pressure in bounce cylinder
d′	—resistance at inner dead center, to match bounce energy as per curve c′

bounce cylinders would be quite sufficient to permit the safety device to respond in time.

The time required for the bounce cylinder pressure to adapt itself to a new position of the outer dead center can likewise be read off the stroke diagram, as Figure 11 shows, where the shorter strokes last through several cycles. The fuel injection was reduced at point A by about one-third. The inner dead center first shifts outward, but after about six cycles it returns practically to its former position, thus indicating that the pressure adjustment in the cushion is complete. The time necessary for this adaptation was about 0.6 second. A displacement of the position of the inner dead center of the same order of magnitude but of inverse direction is shown between points C and D when the fuel injected has again been increased to its original value.

ENERGY DIAGRAM WITH MAXIMUM FRICTION

The effect of friction on the operation of gasifiers can likewise be studied with the aid of energy diagrams. Thus, on the graph in Figure 10 are shown the operating conditions with maximum friction above which operation is no longer possible, even if the governor controlling the fuel injection pumps were to open them to the maximum permitted by the cam mechanism, let us say to $\lambda = 1.8$.

The diagram of operation with maximum friction A′-B′-C′-D′ reveals that the gasifier can operate no longer if the mechanical losses attain a value above five times their normal value, i.e. 3700 kg m per meter of stroke. This property of free piston engines constitutes a very efficient protection against serious damage since even comparatively slight scoring or friction stops the machine before serious damage has been produced. In this case, the lack of a flywheel is an advantage compared to engines of classical design where the energy stored in the flywheel often aggravates the damage.

In addition, it is highly improbable that the friction will increase excessively, for the least variation in mechanical losses is indicated instantly by the stroke recording indicator.

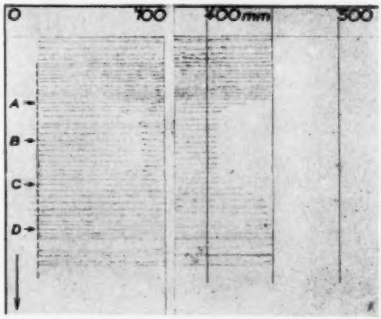

Figure 11.—Stroke Diagram with Prolonged Displacement of Outer Dead Center

In A—fuel charge reduced about 30%
In B—inner dead center restored to initial position
In C—fuel charge increased by 30%
In D—inner dead center restored to initial position

OPERATION WITH DEFECTIVE VALVES

Among other troubles which may arise and are easily recognizable on test, we may mention the failure of the automatic intake and discharge valves of the compressor. Tests were made by connecting the scavenging air receiver to the compressor cylinder by a duct provided with a gate valve on the one hand and the compressor cylinder to the ambient air on the other. In this way the failure of the intake and discharge valves can be simulated. The flow area of the by-pass duct was 14 cm² (2.17 inches²) representing a considerable part of total flow which could exist only when several valves have failed or were badly worn. The stroke diagram, Figure 12, shows that the effect of such a leak, even though it is a considerable one, is scarcely perceptible. On these two diagrams the piston strokes were first recorded with the by-pass gate valve closed. At point A, the gate valve between the receiver and the compressor cylinder was opened suddenly; at point B, the gate valve between the compressor cylinder and the ambient air was also opened rapidly. In both cases the leakage was practically without effect on the operation of the gasifier. These machines apparently could stand considerably greater leaks without deleterious results.

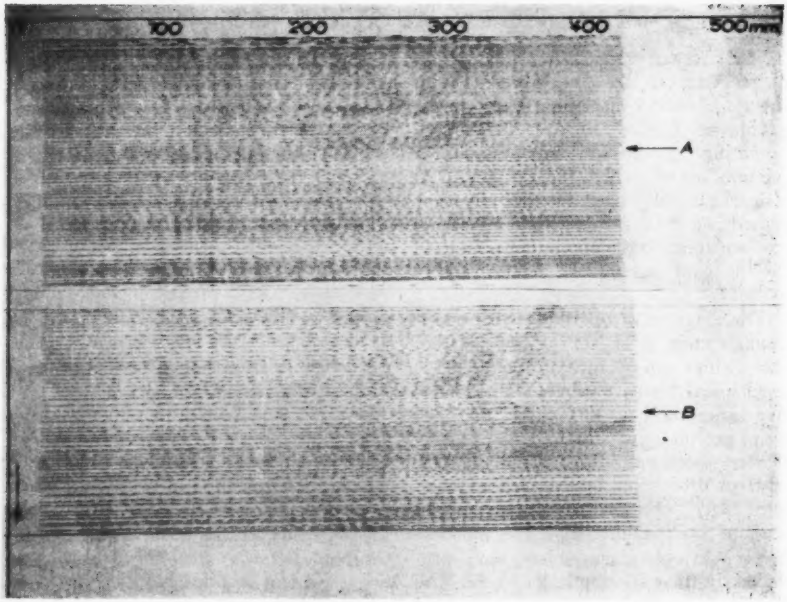

Figure 12.—Stroke Diagram
A—leak receiver to compressor cylinder
B—leak compressor cylinder to atmosphere

The above study of operational stability shows that free piston gasifiers will operate with perfect reliability and uniformity despite the absence of crankshaft restraint. Collision of the two power pistons or impact between the compressor pistons and cylinder heads is absolutely impossible. Even in the case of temporary disturbances due to faulty fuel injection, excessive friction, or leaky valves, normal operation undergoes scarcely any change whatever. On the other hand, the elimination of the classical crank mechanism results in great flexibility of operation and easy adaptation to the variations of output of the turbine fed by the gasifier.

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NAVAL SHIPYARD COST ACCOUNTING

CAPTAIN EMMETT E. SPRUNG, U.S.N.

THE AUTHOR

CAPTAIN SPRUNG graduated from the United States Naval Academy in 1923. As a member of the Construction Corps until its amalgamation with the line, since when he has been designated as an Engineering Duty Officer, Captain Sprung acquired wide experience in practical naval accounting both in Naval Shipyards and in the Navy Department. This article was suggested by a member and was kindly prepared by Captain Sprung at the request of the Editor. It is presented as a basic document in a more popular and understandable form than in the official manuals. It is basic because after 40 years there are impending changes and this article describes the base on which any such changes will be made.

It seems advisable to preface this article by reference to the dictionary. In Merriam-Webster Unabridged, I find these definitions, the italics being mine.

ACCOUNT . . . Also the *series of items occurring under any one heading* in the ledger of a business concern, or the final balance due on such a series.

ACCOUNTING . . . The art or system of making up or stating accounts; the body of scientific principles underlying the keeping and explanation of business accounts.

COST ACCOUNTING . . . Any *system of accounts which reveals the cost elements of material, labor and overhead* incident to the production and distribution of a commodity or the rendering of a service; also the keeping of such accounts.

I thus understand from these definitions that an account is but a classification of money into a single and separate relationship series. I also understand that cost accounting is a *system of accounts* (plural) whose purpose is to reveal the cost elements of labor, material and overhead.

Now most everyone understands what constitutes labor and material, but what is overhead? Again, let us go to the dictionary.

OVERHEAD . . . those *general charges of expenses or expenses collectively* in any business *which cannot be charged up as belonging exclusively to any part* of the work or product as rent, taxes, insurance, lighting, heating, accounting and other office expenses and depreciation.

It is interesting to see the Navy's definition of overhead. Referring to paragraph 67000 of the Bureau of Supplies and Accounts Manual, we have: "Total overhead expenses include all costs that are not charged directly to productive work. Examples of those

costs are depreciation, disability, direct expenses of productive shops, drydock and marine railway expenses and maintenance and operating expenses of the general expense functions." It is thus seen that, except for the different terms used, which are distinctive within the Navy, the definition is substantially identical with that given by Merriam-Webster.

The title of this article is "Naval Shipyard Cost Accounting." But since by definition cost accounting is but a system of accounts to reveal certain cost elements, then in order to discuss the system I must first discuss the elements of the system, that is, accounts.

By definition an account is a classification of money transaction and in the National Budget we find large classifications identified to the various branches of the Government, such as to the Department of Agriculture; to the Department of Commerce; and among others, to the Department of the Navy. In the funds coming to the Navy I find other classifications. The Navy knows these by the titles "Appropriations" or "Funds." Thus we have Appropriation, Maintenance, Bureau of Ships; Maintenance, Supplies and Accounts; and many others. A cursory count in the Supplies and Accounts Manual lists some 200-odd appropriations or funds. Some of these classifications are for different fiscal years, currently of fiscal years 1948, 1949 and 1950. Some of these appropriations are not restricted to a fiscal year but are continuing, presumably until the funds appropriated thereunder are expended, or authority to expend is voided by an Act of Congress, or until the objective of the appropriation is completed.

The organization of the Department of the Navy into Bureaus and Offices is basically a functional type of organization. So it is not surprising to find that certain appropriations are associated with, or fall under the cognizance of single Bureaus or Offices. Thus appropriation 1701002, Medical Department, Navy 1950, is assigned to the Bureau of Medicine and Surgery as the cognizant Bureau. 1701601, Maintenance, Bureau of Ships, 1950, goes to the Bureau of Ships; 1790307, Operation and Conservation of Naval Petroleum Reserves, 1949, goes to the Office of the Secretary, and so on for each and every appropriation or fund.

Each of these accounts or appropriations has sub-classifications or sub-accounts. These are known as "Expenditure Accounts." Such accounts serve to classify all funds expended according to the end-use or purpose for which such expenditures are made. The reason for so classifying expenditures is to provide a detailed analysis of the cost of maintaining and operating the Navy Department and the Naval Service. Such analysis is required in order to accomplish the following:

1. Effective administrative control of the purposes for which appropriations and funds are expended.
2. Preparation of budget estimates which correctly reflect the requirements of the Navy Department and the Naval Service and which can be justified successfully before Congress.
3. Preparation of statistical reports required by Congress showing the purpose for which appropriation of funds have been extended.
4. Comparison of the cost for performing certain work at a Naval activity with the cost of having it done at a private concern.
5. Comparison of the cost for performing work between activities within the Naval establishment.

The entire family of expenditure accounts is a broad one and is so designed to cover any type of end-use expenditure. Within the family are accounts for ship repair, ship alterations, ordnance, etc. For the purpose of this article the

most significant expenditure accounts fall into the 44000 and 45000 Series. In those series nearly all overhead charges are classified. It will be profitable to examine these series in somewhat greater detail.

Referring to the Bureau of Supplies and Accounts Manual, paragraph 72219, the items chargeable under 44000 Series, Plant Maintenance, Upkeep and Repairs, are as follows:

1. Maintenance, upkeep and repairs of land, buildings and improvements, and equipment, which do not result in increasing the value of the property. I digress here to state that should it increase the value of the property, such costs are charged to plant account, as a fixed asset, or capital expenditure.
2. Maintenance, Upkeep and Repairs of Miscellaneous Equipment, Plant Appliances, Machinery and Machine tools, portable Power Tools and Loose and Hand Tools.
3. Replacements and additions of Miscellaneous Equipment, Plant appliances, Machinery and Machine Tools, Portable Power Tools, and Loose and Hand Tools, when the cost is less than $50.00. I again digress to state that when the cost is more than $50.00, it will be charged as a plant account item.

In the 45000 Series, Plant Operations, I find those items chargeable:

1. To this Series is charged the cost of miscellaneous labor and consumable supplies, losses and gratuities, and supervision.

I do not want to leave the impression that there are no classifications of expenditures other than expenditure accounts. Most of the other classifications, which I will explain but briefly, do not fall within the purview of this article. Hence, they will be but identified and not discussed in detail.

(a) All expenditures are classified to individual vessels or stations by accounting numbers. Thus expenditures at a particular station or by a particular vessel can be so identified.

(b) The Bureau of the Budget requires that all expenditures be classified as to Object. This classification is based upon the nature of the services, articles or other items for which funds are expended, as distinguished from the purposes for which such expenditures are made. Thus expenditures for personal services, supplies and materials are classified as such irrespective of the end-use, which may be manufacture of equipment, or erection of structures, or repair of ships. These Object classifications are:

Personal services	01
Travel	02
Transportation of things	03
Communication services	04
Rents and utility services	05
Printing and reproduction	06
Other contractual services	07
Supplies and materials	08
Equipment	09
Land and structures	10
Grants, subsidies, and contributions	11
Pensions, annuities, and insurance losses	12
Refunds, awards, and indemnities	13
Interest	14
Public debt retirement	15
Investment and loans	16

(c) Certain expenditures are classified according to property classes, such expenditures being characterized as expenditures for plant or capital purposes. These classifications have been established in order to provide a systematic recording of plant property and in order to segregate such property according to varying rates of depreciation. All plant prop-

erty falls within one of three property classes and when acquired by purchase, transfer, issue from store, or by any other means, shall be taken up in the plant account under the applicable classes which are as follows:

(1) Land in property class 1.
(2) Building and improvement in property class 2.
(3) Equipment in property class 3.

All classifications previously discussed are applicable Navy wide, applying to all stations. While the details of the present forms of classifications are of somewhat recent origin, the basic system of expenditure classifications has been in effect for many years.

In certain Naval stations, we find large scale industrial institutions which, according to the dictionary definition, may be said to be engaged in "the production and distribution of a commodity or the rendering of a service."

For management purposes, the officials of such installations, in the Navy as in commercial life, require knowledge in detail of the elements of material. labor and overhead that enter into the cost of production of output, be it a commodity or a service. Such activities have further accounting practices in use known as Industrial Cost Accounting. All Naval Shipyards and the Gun Factory at Washington, D. C. are activities operating under the Industrial Cost Accounting System.

The origin of this Industrial Cost Accounting traces back to the year 1909 when President Taft took office. His Secretary of the Navy was named as The Honorable George L. von Meyer. Shortly after he took office, the Secretary engaged a firm of certified public accountants, the Marwick, Mitchell and Company, to study the industrial activity of the Navy and install an Industrial Accounting System. This firm of accountants took as their hypothesis that the Navy Yards (as they were then called) were industrial plants, and that the Bureaus and ships were their customers. Work that came to a Navy Yard in the form of Work Requests from the different departments of the ship, or from Bureaus, was regarded as productive work. Work in the nature of plant betterments, and work in commercial lines which would normally be done at a charge against the depreciation reserve was also regarded as productive work. Operating expenses of the Yards not directly allocable to productive work, and small plant repair jobs estimated to cost less than $100, were to be regarded as overhead expenses.

Overhead expense was divided into three major classes—General Expense items which embraced General Administration, Yard Transportation Systems, Sewage Systems, Water Systems, etc.; Power expense which included power plant buildings, repairs to machinery, distribution expenses of power, etc.; and Shop Expense, which was further divided into as many parts as there were shops, with such items as supervision, minor repairs to shop buildings; repairs to shop machinery and shop equipment and other operating materials such as oil, grease and waste, each identified as to cost. This Industrial Cost Accounting System as then placed in effect in 1909 by the firm of accountants is still in effect with many, many modifications, but no significant alterations having been made in the following years.

The cost element of direct labor and direct material going into a unit of output of an industrial plant is relatively easy to determine, being simply that labor or material which can be identified as being used solely, directly, and exclusively to produce that unit of output. It is in the determination of the element of overhead cost going into the cost of output that the most difficulty is

experienced. Most of the remainder of this paper will be restricted to the study of overhead costs. Before proceeding further, it is necessary to give a discussion of the job order system in use at Naval Industrial activities.

To expend funds from any appropriation requires first an expenditure authorization. Such authorizations come from cognizant Bureaus or Offices, sometimes direct to the activity, and sometimes through devious Command channels. But irrespective of the paths used in reaching the Industrial Yards, such authorizations all extend originally from the cognizant Bureau or Office.

Authorizations are received in the form of an allotment or project order for the accomplishment of work, or sometimes by a requisition covering the purchase of material.

Many rules exist to cover means of determining whether an allotment or a project order is the appropriate authorization in a specific case. Distinctions can be made broadly on the following basis:

(a) An allotment is used to authorize the expenditure of funds during the first year of an appropriation's life, and is used for work having continuing recurring characteristics, to thus finance that work which will be accomplished in the fiscal year in which the authorization is issued. Thus ship repair in fiscal year 1950 is authorized by Fiscal year 1950 Allotment, and correspondingly the Ship Repair Program in Fiscal Year 1951 will be authorized against a Fiscal Year 1951 allotment.

(b) A project order is an order for work or material placed with a naval activity under the authority of the Act of July 1, 1922 and the Act of June 2, 1937. It serves to obligate funds in the same manner and to the same extent as a contract with a party outside the Naval Establishment. It is issued to cover work or material of a non-recurrent nature, as distinguished from routine maintenance or operation, and there will be an intention that the work there named will be commenced within a reasonable length of time. It must contemplate personnel services beyond those incidental to contract procurement on the part of the activity to which issued. It nearly always covers work that will not be completed within the first year of an appropriation's life.

(c) Requisitions authorize the purchase of material and, with but few exceptions, when issued in Naval Shipyards, are chargeable to allotments or project orders held at the Yards, and are for the purpose of obtaining material to accomplish work authorized by the allotment or project order. Many other types of requisitions are issued, particularly by Bureaus and Offices, but for the purpose of this article, this incomplete definition will perhaps suffice.

When a work authorization is received by a Naval Shipyard, a "job order," identified by its own distinctive serial number is issued. This job order, besides defining the allotment or project order ultimately chargeable with the cost of the work authorized, defines the work to be done, as to—

(a) What is to be done—to manufacture six valves, or to repair a pump, etc.

(b) How the work is to be done—that is, the engineering specifications, the kinds of materials to be used, the processes to be used.

(c) Who shall do the work—what productive shops or trades are involved—that is, machinists, sheetmetal workers, etc.

The job order is the total work specifications. It limits the work to so much and no more, but also requires not less than a specified amount of work. It outlines material specifications and, to the extent required, it details production processes and procedures.

All labor time cards of men working directly on the work specified by the job order are marked with that job order serial number, and the cumulative total becomes the direct labor cost of the work. In a like manner, all material stubs for material used directly on the work authorized by the job order, are marked with that job order serial number, and, being accumulated, become the direct material cost of the work. Commercially these are comparable to direct charges.

There remains the determination of the element of overhead that should be charged to that particular job order as the part of the final cost. The overhead element is costed to the job order as a prorated percentage of the direct labor charges. It is in the determination of what percentage of prorate to apply, and what it represents, where difficulty arises.

Previously, I have stated that overhead charges (known commercially as indirect charges), when made, are classified in the 44000 or 45000 series of expenditure accounts. For management purposes, however, it is not enough to know that a certain charge is overhead and therefore classified in certain expenditure accounts. Such information may be sufficient from a fiscal point of view, or for budgeting purposes, but the management team in a shipyard, in order to properly control the magnitude of overhead cost must know in far greater detail the specific purpose of such charge.

Therefore, overhead expenditures are sub-classified to each productive shop and to each division of the General Expense functions. Even these sub-classifications are not in sufficient detail for good overhead cost control. Hence, we find, in the productive shops, an even further sub-division of overhead expenses into what is known as cost classes.

Before going further, I must define a Productive Shop and the General Expense functions.

A productive shop is an organizational unit primarily and directly engaged in the manufacture, alteration, modification, overhaul and repair of naval vessels, and their component items of equipment. The labor force in each productive shop generally consists of a group of specific tradesmen such as electricians or machinists, etc. This labor force may perform its work either within a shop building proper, or outside the shop on ships, or other places. When their work is of a productive nature, creating the normal output of the Yard, the cost becomes a direct charge to specific output job orders. When the work is of a non-productive nature, or impractical to identify to a specific job, the cost thereto is charged to maintenance and operation accounts and as such becomes a portion of the industrial overhead to be costed by percentage prorate to job orders representing the output of the Yard.

Independent of the Productive Shops, there are nine General Expense functions, namely:

000	Administration
100	Supply System
200	Transportation
300	Power generation and distribution
400	Communications Systems and Distributing Systems other than power
500	Ground and ground structures
600	Waterfront and waterfront structures
700	Personnel facilities
800	Hospitals, medical centers, medical research commands, medical department schools, medical supply depots, and dispensaries.

It is important to know what these groups contain. Their title gives us a hint. The exact items can be found in Supplies and Accounts Manual, paragraph 72324. For the purpose of this discussion I will list a few of the more major elements in each group:

000—Administration
- Accounting and disbursing offices
- Planning and estimating
- Drafting room
- Progressmen
- Central industrial officer
- Central toolroom

100—Supply System
- Supply Office
- Material purchasing, receiving, records, requisitions, processing, etc.

200—Transportation
- All the equipment commonly associated with the transportation job and the personnel in shipyard Shop 02 having to do with the operation, repair, and maintenance of that equipment, irrespective of its character, including railroad, automotive, water.

300—Power
- The generation and distribution of electric power
- Fresh and salt water
- Gas
- Electricity
- Steam
- Hot water heat

400—Communication Systems and Distributing System other than power
- Radio Stations
- Telephone
- Air Conditioning
- Refrigeration

500—Grounds and ground structures
- Bridges
- Farms
- Grounds
- Fences
- Street Lighting

600—Waterfront and waterfront structures
- Piers
- Quay wall
- Drydocks
- Marine railway
- Wharves

700—Personnel facilities
- Bachelor Officer quarters
- Enlisted men's barracks
- Chapel
- Marine barracks
- Officers' quarters

800—Hospitals
- Medical centers
- Medical research commands
- Medical department schools
- Medical supply depots
- Dispensaries

A point: First Aid dispensaries in the industrial part of the Yard, as distinct from the main dispensary, is a charge in the 000 Group, rather than this group.

As stated, sub-classification of overhead expenditures is made to each productive shop, for that productive shop, and to each division of general expense for that division of the general expense function. Overhead expenditures in each productive shop are further sub-classified into cost classes.

List of Productive Shop Cost Classes —The appropriate cost class number shown is used as a suffix to the applicable expenditure accounts in the 44000 and 45000 series.

Miscellaneous Labor and Consumable Supplies
- Miscellaneous labor and consumable supplies 01
- Office, shop store, and tool room labor and supplies.....02
- Fuel 03
- Safety 06

Losses and Gratuities
- Sick leave 10
- Damage and waste.....11
- Scrap credits 12
- Time allowed 14
- Instruction of employees.....15

Travel time (to be used ordinarily only by Pearl Harbor)...16
Annual leave 17
Holiday leave 18
Land and Appurtenances
Land and appurtenances.....20
Buildings and Structures
Buildings and structures.....30
Miscellaneous Equipment
Miscellaneous equipment 40
Office furniture 41
Plant Appliances
Plant appliances 50
Machinery and Machine Tools
Machinery and machine tools (including motors and shafting) 60
Portable Power Tools
Portable power tools.....70
Loose and Hand Tools
Loose and hand tools.....80
Supervision
Supervision 90

The following expenses are chargeable to the Individual Productive Shop Cost Classes:

1. Miscellaneous Labor and Consumable Supplies
   (a) Miscellaneous Labor and Consumable Supplies (01). This class will be charged with the cost of the following:
      (1) Labor incident to operation which cannot be allocated as direct labor to a specific job order; this includes labor involved in cleaning shops and building, attending power plant, moving material in shops, and attending pump well;
      (2) labor and supplies incident to testing of machinery and appliances;
      (3) labor and material for periodic qualification tests of rated welders;
      (4) supplies and materials incident to operation which are of such general uses and in such small quantities as not to permit charging to specific job orders; this includes oil, brooms, brushes, toilet paper, ice, etc.;
      (5) for drydocks, cost of cleaning of dock and caisson not incident to docking ships.
   (b) Office, Shop Store, and Tool Room Labor and Supplies (02). This class will be charged with the cost of labor and material expended in connection with the operation of the offices, shop store rooms, and tool rooms, such as pay of clerical employees, pay of employees in storerooms and tool rooms, supplies such as soap, stationery, office supplies, floor sweep materials, etc.
   (c) Fuel (03). Fuel will be charged with the cost of all fuel, fuel oil, and power consumed, including electricity, steam, gas, etc.
   (d) Safety (06). This class will be charged for all expenses incurred in connection with accident prevention and safety engineering which cannot be allocated to a specific job order.
2. Losses and Gratuities
   (a) Sick Leave (10). This class will be charged with the pay of all employees on authorized sick leave.
   (b) Damage and Waste (11). This class will be charged with the following:
      (1) damage and waste for defective work of any character (foundry, forge, machine shop, etc.);

      Note: Authority for charging must be obtained from the supervisor of the

force making the error, or a mutual superior of the supervisors concerned.

(2) minor discrepancies or omissions in charges, by the accounting department, the shop supervisor being notified.

(c) Scrap Credits (12). This class will be credited with the value of excess material, machinings, and short ends, which have been taken up on Returned Material Credit Memorandum when the specific job order for which the material was originally issued cannot be determined.

(d) Time Allowed (14). This class will be charged with time lost due to breakdowns, cleaning machines, waiting for materials, tools, drawings, and plans, and other time between the finishing of one job and the starting of another; also for pay of employees taking examinations.

(e) Instruction of Employees (15). This class will be charged with the following:

(1) time connected with instruction of civilian employees when not practicable to charge a specific job order;

(2) time of civilian apprentices attending apprentice schools, the charge being made under the proper producing section;

(3) time of apprentices periodically assigned to drafting rooms for instruction when not practicable to charge a specific job order, charge being made under the proper producing section;

(4) labor and material for special instruction or training of rated welders to develop ability to make satisfactory welds on work of unusual character, such as armor plate, duraluminum, or some other new alloy;

(5) instruction or training of student employees (apprentices or other) in welding.

Note: Labor and material expended for experimental welding when not designated as a qualification test, or when not designated for instruction and training of student employees (apprentices or other) and rated welders will be a direct charge to a specific job order.

(f) Travel Time (16). This class is to be used ordinarily only by Pearl Harbor. This cost class shall be charged with travel time allowed employees reporting for work or when carried on the payroll at termination of their services, or when time is allowed for travel going on or returning from leave.

(g) Annual Leave (17). This class will be charged with the pay of employees on authorized annual leave.

(h) Holiday Leave (18). This class will be charged with the pay of employees on authorized legal holidays but receiving compensation.

3. Land and Appurtenances (20). This class will be charged with the cost incurred in connection with the improvement or maintenance of all land and grounds including improvements except improve-

ments in connection with acquisition (dredging, etc.), such as sea walls, piers, roads, walks, gutters, lawns, underground conduit systems, railroad tracks, distributing systems outside of buildings, sewer and water systems, and drains.

4. Buildings and Structures (30). This class will be charged with the cost of the following:
   (1) repairs in connection with shops, such as repairs to oil, gas, water, and power distributing systems within the shop;
   (2) repairs to buildings and structures, including roofs, floors, walls, railroad tracks, crane railways and shop overhead traveling cranes in buildings, drains, inside sewers, inside piping (air, oil, gas, steam, water), radiators, and inside wiring;
   (3) repairs to body, caisson, and permanent fittings of docks, repairs of pumpwell buildings, hulls of floating docks, marine railway structures, and dock latrines.

   Note: Blocking damage caused by an individual ship will be charged to the job order for docking the ship.

5. Miscellaneous Equipment
   (a) Miscellaneous Equipment (40). This class will be charged with the cost of repairs to articles other than plant appliances which assist in work, and with the cost of replacements and additions of such articles when the cost is less than $50. These items include furniture (other than office), work benches, handcarts, trucks, time clocks, hand hoists, templates and patterns (purchased for standard use), molds, flasks, nonfiction books, etc.

   Note: This class will not be charged with labor and material expended in preparation of drawings for, or manufacture of, patterns, dies, jigs, templates, holding-down devices. When for a standard product, the expenditures will be lodged against a separate specific job order for which an allotment is made or funds are provided under a specific appropriation. When for a non-standard product, the expenditures will be lodged against the job order concerned.

   (b) Office Furniture (41). This class will be charged with the cost of repairs to office furniture, and with the cost of replacements and additions when the cost is less than $50. Office furniture includes desks, file cabinets, office chairs, bookcases, desk lamps, etc.

   Note: This class does not include typewriters, adding machines, calculators, and mimeograph equipment which are included in class 50, plant appliances.

6. Plant Appliances (50). This class will be charged with the cost of repairs to plant appliances, and with the cost of replacements and additions of such appliances when the cost is less than $50. Plant appliances include the following items:
   (1) power plant appliances specifically classed as individual items of plant;
   (2) power plant boilers, piping, furnaces, all railroad rolling stock, vehicles, derricks, shears, yard boilers and engines, bicycles, automobiles, yard scales, all floating property not on Navy list, yard or shop cranes (other than shop

overhead cranes), and test boards;

(3) typewriters, calculating machines, addressographs, blueprinting machines, mimeograph equipment, and other items which may be specifically designated by the Bureau of Supplies and Accounts;

(4) capstans, winches, and floats other than floats available for general water front use.

7. Machinery and Machine Tools (Including Motors and Shafting) (60). This class will be charged with the cost of repairs to machinery and machine tools, and with the cost of replacements and additions of such items when the cost is less than $50. Machinery and machine tools include the following items:

(1) stationary tools using power or fuel in operation, such as bending rolls, boring mills, carving machines, drill presses, grinding machines, lathes, milling machines, planers, punch presses, sandpapering machines, saws (band, circular, and jig), steam hammers, trip hammers, mortise machines, blowers cupolas, forges, furnaces, etc.;

(2) power plant machinery, such as engines, generators, etc.;

(3) pumping plant and caisson machinery, including suction and discharge pipes inside and outside of pump well, pumps, valves, motor, etc.

8. Portable Power Tools (70). This class will be charged with the cost of repairs to portable power tools, and with the cost of replacements and additions of such items when the cost is less than $50. Portable power tools include those portable tools using power or fuel in operation, such as air tools, portable electric drills, gasoline torches, riveting forges, portable acetylene welding apparatus, etc.

9. Loose and Hand Tools (80). This class will be charged with the cost of repairs to loose and hand tools, and with the cost of replacements and additions of such items when the cost is less than $50. Loose and hand tools include tools not requiring fuel or power for use and tools of a type usually stored in a tool crib. This cost class will not be charged with labor and material expended in preparation of drawings for, or expended in the manufacture of, patterns, dies, jigs, templates, and holding-down devices. When for a standard product, the expenditures will be lodged against the job order concerned unless the bureau or bureaus interested direct that the expenditures be lodged against a separate specific job order for which an allotment is made or funds are provided under a specific job order for which an allotment is made or funds are provided under a specific appropriation. When for a nonstandard product, the expenditures will be lodged against the job order concerned.

10. Supervision (90). This class will be charged with the wages of all master mechanics, all foremen, all quartermen, and all leading men.

I expect, by this time, if I have accomplished anything, I have established in the reader's mind that expenditures in Naval Shipyards are classified, multiple-classified, reclassified, and sub-classified many times.

So far I have avoided stating what appropriations pay, or are charged with, the overhead expense. Such a question, it will be appreciated, is independent of the cost of work, being only a question of who pays the bill. In practice overhead costs are paid for by several appropriations, but principally are borne

by the Appropriations "Maintenance Bureau of Ships," "Maintenance Yards and Docks," and "Maintenance Supplies and Accounts." What portion each appropriation pays is determined primarily by the basic cognizance of the Bureaus named, augmented or modified by agreements between Bureaus, long-time practice of charging certain items to certain Bureaus, or by directives to affected Bureaus, received from higher authority.

I now propose to introduce an exhibit or chart which I have personally originated. This chart generally attempts to show—

(1) what makes up overhead expenditures;
(2) what makes up total expenditures;
(3) who pays for what overhead expenditures;
(4) the total industrial cost of work.

Examining Exhibit 1, the two blocks on the upper left should be fully understandable in the light of my foregoing discussion, since they simply represent direct labor and direct material to productive work. These costs will be paid by the appropriation of the Bureau or Office ordering the work.

Exhibit 2 is actual return of a typical productive shop, a sheetmetal shop, for a typical shipyard for the month of October 1947. It shows the direct shop expense in that productive shop. Similar reports are made for all productive shops and, when combined, a recapitulation is obtained governing the overhead expenses in all productive shops. Exhibit 3 is a recapitulation of the returns from all productive shops of the same shipyard for the same month, October 1947. Confining attention to lines 1 to 25 of Exhibit 3, the total of these lines represent the third block on the upper left of my chart. I call attention to the cost class numbering shown on each line of Exhibits 2 and 3. I also call attention to the fact that line 17, cost class 30, is the "Maintenance, Yards and Docks," portion of the direct shop expense shown on Exhibit 1, the remainder of direct shop expense all falling to "Maintenance, Bureau of Ships" for payment.

The remaining nine blocks, including the power block near the center of Exhibit 1, goes to make up the general expense portion of overhead. I have indicated on Exhibit 1 what Bureau, or appropriation, pays what portion of this general expense.

Lines 26, 27 and 29A on Exhibits 2 and 3 need explanation. The industrial costs of power used by productive shops, is distributed to the various shops on the basis of predetermined formulae. This is known as the power transfer, and is so shown and labeled in Exhibit 1. In a like manner the General Expense transfer is distributed to the various productive shops, and is shown and labeled on Exhibit 1 as the General Expense transfer.

The net leave accrued, shown on line 29A of Exhibit 2 and 3, is simply the difference, plus or minus, between the leave actually taken and paid to the men of the shop, cost class 17, and the leave earned by the men of the shop. Thus, limiting of annual leave in periods of high work load, or conversely, the granting of considerable leave in slack work load periods, has no effect on the overhead rates.

Several points on Exhibit 1 need further explanation. You will note that depreciation and disability costs are apparently calculated, and then after calculations are made, are seemingly not used. This is actually true in Naval Shipyards. There can be no dispute that both items are part of true industrial costs and hence should theoretically be included. But because of the special peculiarities of government financing, it is more convenient not to so include them in the final cost. Neither represent a current appropriation cost. In fact, disability costs are not even paid by naval appropriations, being paid entirely by another appropriation of the government.

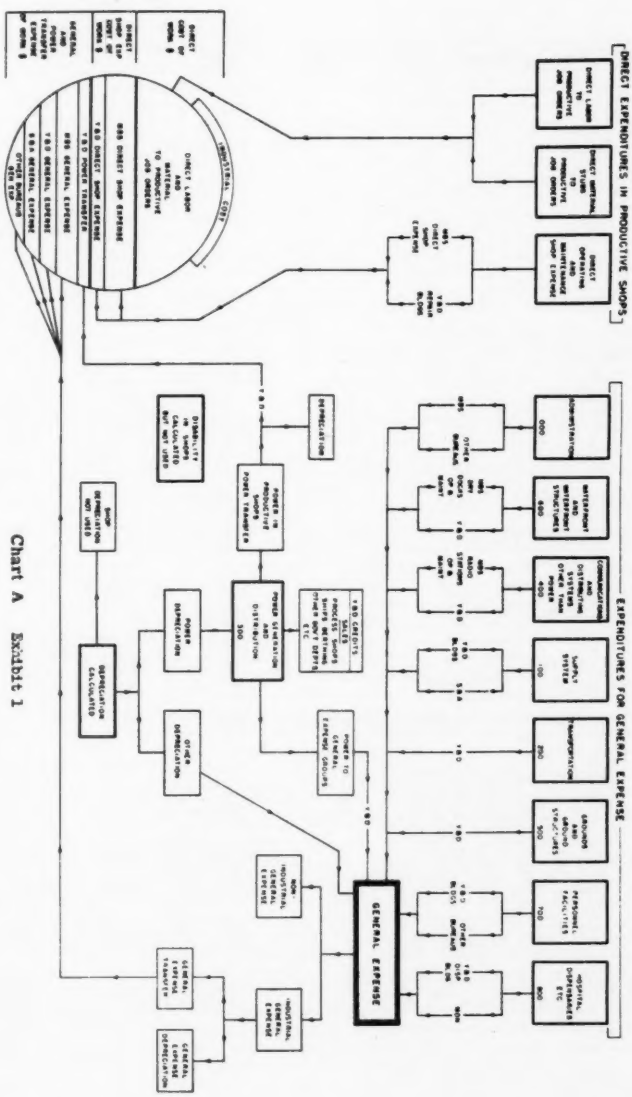

Chart A Exhibit 1

DETAILED STATEMENT OF SHOP EXPENSE
Nav. S. and A. Form 299 (Rev. Jan. 1946)

Shop or Drydock (*Name*)	(*Number*)	Month of
Sheetmetal	17	22 Sept.—26 Oct. 1947
Reporting Activity (*Name*)	(*Number*)	(*Location*)
xxxxxxxxxxxxxxxxxxxxxx	xxxxx	xxxxxxxxxxxxxx

Line No.	*Cost Classes*	*Month*	*Fiscal Year to Date*	*Budget Fiscal Year*
	PLANT OPERATIONS:			
1	01. Miscellaneous Labor	1446.61	5210.97	
2	01. Consumable Supplies	253.67	745.65	
3	02. Office, Shop Store, and Toolroom Labor	2776.19	8730.95	
4	02. Office, Shop Store, and Toolroom Supplies	8.80	45.34	
5	03. Fuel	16.36	26.99	
6	04. Safety		101.48	
7	10. Sick Leave	2299.04	5792.25	
8	11. Damage and Waste	45.00	45.00	
9	12. Scrap Credits	CR 1788.87	CR 4667.50	
10	14. Time Allowed	188.20	1324.93	
11	15. Instruction	1622.47	5760.21	
12	16. Travel Time			
13	17. Annual Leave	12235.75	52513.92	
14	18. Holiday		8861.27	
15	90. Supervision	11090.61	32470.07	
16	Subtotal	30193.83	116961.53	
	MAINTENANCE, UPKEEP, AND REPAIRS:			
17	30. Buildings and Structures	221.45	422.37	
18	40. Miscellaneous Equipment	176.97	434.53	
19	41. Office Furniture	22.88	22.88	
20	50. Plant Appliances	10.41	51.74	
21	60. Machinery and Machine Tools	1120.33	3887.84	
22	70. Portable Power Tools		69.95	
23	80. Loose and Hand Tools	30.84	CR 509.84	
24	Subtotal	1582.88	4379.47	
25	TOTAL DIRECT EXPENSE	31776.71	121341.00	
	OTHER EXPENSE:			
26	xx Power Transferred	1042.26	8248.97	
27	xx General Expense Transferred	42467.92	143780.82	
28	13. Depreciation			
29	19. Disability			
29A	Net Leave Accrued	CR 2985.60	CR 11135.51	
30	Subtotal	40524.58	140894.28	
31	TOTAL EXPENSE	72301.29	262235.28	
	OPERATING DATA:			
32	Industrial Overhead Expense Applied—NSF Job Orders			
33	Industrial Overhead Expense Applied—Other Job Orders	72948.59	267029.75	
34	Over (O) or Under (U) Absorbed Industrial Overhead Expense	O– 647.30	O– 4794.47	
35	Productive Labor—NSF Job Orders			
36	Productive Labor—Other Job Orders	81082.28	275163.44	
37	Productive Hours—NSF Job Orders			
38	Productive Hours—Other Job Orders	61947	214545	
39	Docking Hours			
40	Actual Overhead Rate	89%	95%	
41	Industrial Overhead Rate Used This Month	90%		
42				
43	Special Appropriation Adjustment Rate—Actual			
44	Special Appropriation Adjustment Rate Used			
45				
46				
47				
48				
49				
50				
51				

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EXHIBIT 2

RECAPITULATION—PRODUCTIVE SHOPS

DETAILED STATEMENT OF SHOP EXPENSE
Nav. S. and A. Form 299 (Rev. Jan. 1946)

Shop or Drydock (*Name*)	(*Number*)	Month of 22 Sept.—26 Oct. 1947
Reporting Activity (*Name*) xxxxxxxxxxxxxxxxxxxxxxx	(*Number*) xxxxx	(*Location*) xxxxxxxxxxxxxx

Line No.	Cost Classes	Month	Fiscal Year to Date	Budget Fiscal Year
	PLANT OPERATIONS:			
1	01. Miscellaneous Labor	18188.19	60227.40	
2	01. Consumable Supplies	4860.63	13275.21	
3	02. Office, Shop Store, and Toolroom Labor	12659.60	39362.58	
4	02. Office, Shop Store, and Toolroom Supplies	398.58	1564.96	
5	03. Fuel	2198.69	3638.68	
6	04. Safety	1178.28	5011.00	
7	10. Sick Leave	45741.47	129685.29	
8	11. Damage and Waste	89.25	137.75	
9	12. Scrap Credits	CR 6099.35	CR 21829.24	
10	14. Time Allowed	736.18	6137.23	
11	15. Instruction	20036.69	64802.69	
12	16. Travel Time	5.63	29.57	
13	17. Annual Leave	183359.99	643624.91	
14	18. Holiday	535.69	127859.11	
15	90. Supervision	138414.95	437125.11	
16	Subtotal	422314.47	1510652.25	
	MAINTENANCE, UPKEEP, AND REPAIRS:			
17	30. Buildings and Structures	9716.50	46526.03	
18	40. Miscellaneous Equipment	7533.29	17782.23	
19	41. Office Furniture	91.97	203.94	
20	50. Plant Appliances	4934.09	13522.23	
21	60. Machinery and Machine Tools	16986.48	57123.47	
22	70. Portable Power Tools	990.62	3767.07	
23	80. Loose and Hand Tools	4096.07	11193.01	
24	Subtotal	44349.02	150117.98	
25	TOTAL DIRECT EXPENSE	466663.49	1660770.23	
	OTHER EXPENSE:			
26	xx Power Transferred	9861.72	101228.84	
27	xx General Expense Transferred	655840.25	2249194.64	
28	13. Depreciation			
29	19. Disability			
29A	Net Leave Accrued	CR 65680.79	CR 108531.41	
30	Subtotal	600021.18	2241892.07	
31	TOTAL EXPENSE	1066684.67	3902662.30	
	OPERATING DATA:			
32	Industrial Overhead Expense Applied—NSF Job Orders			
33	Industrial Overhead Expense Applied—Other Job Orders	1020835.93	4135315.99	
34	Over (O) or Under (U) Absorbed Industrial Overhead Expense	U– 45848.74	O– 232653.69	
35	Productive Labor—NSF Job Orders			
36	Productive Labor—Other Job Orders	1252169.28	4297497.54	
37	Productive Hours—NSF Job Orders			
38	Productive Hours—Other Job Orders	964813	3328958	
39	Docking Hours			
40	Actual Overhead Rate	85%	91%	
41	Industrial Overhead Rate Used This Month	82%		
42				
43	Special Appropriation Adjustment Rate—Actual		35%	
44	Special Appropriation Adjustment Rate Used	34%		
45				
46				
47				
48				
49				
50				
51				

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EXHIBIT 3

It will be further noted that the block of total general expense (the heavily shaded block), after being determined, is immediately separated into two portions; namely, Non-Industrial general expense and Industrial general expense, and only the latter carried into the industrial costs. It is recognized that a naval shipyard, besides being an industrial activity, is also a military activity. Many functions are performed in, or by the yard, that have no relationship to the industrial effort of the yard. Naval hospitals and naval receiving barracks, included in the organization of the yard, are excellent examples of functions that have no influence on the industrial output of the yard. The work of the supply activities of a naval shipyard is partly in support of industrial activities of the yard, and partly in general naval supply service to the Navy as a whole. The division of each function of general expense into the two parts is partly by definition, and partly by decisions of a local accounting board convened for that purpose. Thus the 700 and 800 groups of general expense, "Personnel facilities" and "Hospitals" are entirely classified Non-Industrial by definition, while the 100 group, "Supply" system, may be divided 40% Industrial and 60% Non-Industrial by the local accounting board.

Eventually, Exhibit 1 shows all industrial costs flowing to a circle on the lower left. (The segments of this circle are not shown in true perspective). The ratio of overhead costs to total productive labor over a given time; i.e., total overhead cost (this month) over productive labor (this month); or total overhead (fiscal year to date) over productive labor (fiscal year to date) gives the industrial overhead rate, actual, over those periods of time. In theory, this is the prorate percentage to be applied to the direct labor cost of individual job orders (using the rate for separate shops as appropriate to the shops furnishing the direct labor) to thus determine the element of overhead cost that should be costed to that job order. However, since the actual industrial overhead rate is only determinable after the fact, an estimated overhead rate must actually be applied. This estimated rate is established by local accounting boards, from time to time as required, their judgment being guided by the knowledge of past experience and if the estimated rate previously applied has served to absorb or has failed to absorb the actual overhead expenses.

Exhibit 4 is a form of return showing a summary of overhead expense for a typical shipyard for one month.

For those individuals who may wish to carry through the calculations of a typical shipyard for one month, I have included Exhibit 5, the recapitulation of drydock expenses, which when combined with Exhibit 3, the recapitulation of productive shop expense, go to produce the figures on Exhibit 4. I have also included as Exhibit 6, the summarized calculations for the Yard as a whole in establishing actual overhead rates for the Yard as a whole.

To provide better understanding, Exhibit 7, an actual job order, is shown, the back of which is shown as Exhibit 8, the returned industrial cost of that job order, divided into direct labor, direct material, and overhead making up total industrial cost.

For full understanding of this last Exhibit, I must say a word about how "APA" material costs are handled. I presume most of the readers of this article will understand the meaning of "APA." For the benefit of others, I will give a short definition. "APA" material is material that at some previous time has been bought and paid for by certain Naval Appropriations. Thus when "APA" material is used, there is no further appropriation cost. But there is an industrial cost since the material

EXHIBIT 4

Expense for Fiscal Year to Date

16—46936-1 U. S. Government Printing Office

SUMMARY OF OVERHEAD EXPENSE
Nav. S. and A. Form 419 (Revised Dec. 1945)

Yard xxxxxxxxxxxxxxxxxxxxxxxxxxxxxxxxxx
Month 22 Sept. thru 26 Oct. 1947

	Group	Total	Depreciation	Power Distribution	Net Overhead Expense to be Allocated	Nonindustrial		Industrial	
						%	Amount	%	Amount
000	Administration	543959.09	17298.28	4590.44	565847.81	19	107511.08	81	458336.73
100	Supply Systems	167545.77	17005.19	1682.91	186233.87	70	130363.71	30	55870.16
200	Transportation	152431.38	37996.80	1161.92	191590.10	22	42149.82	78	149440.28
300	Power Generation and Distribution	CR 13004.98	48983.97	CR 22786.02	13192.97	100	13192.97	xxx	xxxxxxxxx
400	Communication Systems and Distributing Systems other than Power	96245.50	23769.02	86.47	120100.99	87	104487.86	13	15613.13
500	Grounds and Ground Structures	22653.77	27414.95	1901.70	51970.42	52	27024.62	48	24945.80
600	Waterfront and Waterfront Structures	24611.35	32775.01	—	57386.36	20	11477.27	80	45909.09
700	Personnel Facilities	36149.68	12189.98	1019.65	49359.31	100	49359.31	xxx	xxxxxxxxx
800	Hospitals, etc.	6743.11	412.52	149.70	7305.33	100	7305.33	xxx	xxxxxxxxx
	Total—General Expense Functions	1037334.67	217845.72	CR 12193.23	1242987.16		492871.97		750115.19
900	Productive Shops, Drydocks, and Marine Railways	483235.97	70554.79	12193.23	565983.99	xxx	xxxxxxxxx		565983.99
900	Process Shops	21371.70	5566.55	xxxxxxxxx	26938.25		26938.25	xxx	xxxxxxxxx
	Subtotal	1541942.34	293967.06	xxxxxxxxx	1835909.40	xxx	519810.22	xxx	1316099.18
	Depreciation (S. and A. Form 276)	293967.06	CR 293967.06	xxxxxxxxx	xxxxxxxxx	xxx	xxxxxxxxx	xxx	xxxxxxxxx
	Disability	8765.18	xxxxxxxxx	xxxxxxxxx	8765.18	xxx	xxxxxxxxx	xxx	8765.18
	Net Leave Accrued	CR 65680.79			CR 65680.79				CR 65680.79
	Total Overhead Expense	1778993.79	xxxxxxxxx	xxxxxxxxx	1778993.79	xxx	519810.22	xxx	1259183.57

Net Leave Accrued	65680.79
Total Industrial Overhead Expense Applied	1324864.36
Over or Under Absorbed Industrial Overhead Expense for Month	
Over or Under Absorbed Industrial Overhead Expense for Fiscal Year to Date	

Abstracted Public Vouchers included in above_____

REMARKS

EXHIBIT 4

16—46936–1 U. S. Government Printing Office

DETAILED STATEMENT OF SHOP EXPENSE
Nav. S. and A. Form 299 (Rev. Jan. 1946)

Shop or Drydock (*Name*) (*Number*)	Month of
Recapitulation Drydocks Marine Railways	22 Sept.—26 Oct. 1947
Reporting Activity (*Name*) (*Number*)	**(*Location*)**
xxxxxxxxxxxxxxxxxxxxxx xxxxx	xxxxxxxxxxxxxx

Line No.	*Cost Classes*	*Month*	*Fiscal Year to Date*	*Budget Fiscal Year*	
	PLANT OPERATIONS:				
1	01. Miscellaneous Labor	3070.95	8480.76		(1)
2	01. Consumable Supplies	169.92	449.13		(2)
3	02. Office, Shop Store, and Toolroom Labor	484.49	919.28		
4	02. Office, Shop Store, and Toolroom Supplies	23.78	23.78		(3)
5	03. Fuel		126.23		(4)
6	04. Safety		280.72		(5)
7	10. Sick Leave				
8	11. Damage and Waste				(6)
9	12. Scrap Credits				(7)
10	14. Time Allowed				
11	15. Instruction				
12	16. Travel Time				
13	17. Annual Leave		305.54		(8)
14	18. Holiday		202.45		
15	90. Supervision	1442.91	2418.30		(9)
16	Subtotal	5192.05	13206.19		(10)
	MAINTENANCE, UPKEEP, AND REPAIRS:				(11)
17	30. Buildings and Structures	10323.71	27987.93		(12)
18	40. Miscellaneous Equipment	732.24	5086.40		(13)
19	41. Office Furniture				
20	50. Plant Appliances	116.76	251.51		
21	60. Machinery and Machine Tools	204.92	335.21		
22	70. Portable Power Tools	2.80	2.80		
23	80. Loose and Hand Tools		189.99		(14)
24	Subtotal	11380.43	33853.84		(15)
25	TOTAL DIRECT EXPENSE	16572.48	47060.03		(16)
	OTHER EXPENSE:				
26	xx Power Transferred	561.94	3694.65		
27	xx General Expense Transferred				
28	13. Depreciation				
29	19. Disability				
30	Subtotal	561.94	3694.65		
31	TOTAL EXPENSE	17134.42	50754.68		
	OPERATING DATA:				
32	Industrial Overhead Expense Applied—NSF Job Orders				
33	Industrial Overhead Expense Applied—Other Job Orders	9340.00	35550.00		
34	Over (O) or Under (U) Absorbed Industrial Overhead Expense	7794.42	U- 15204.68		
35	Productive Labor—NSF Job Orders				
36	Productive Labor—Other Job Orders				
37	Productive Hours—NSF Job Orders				
38	Productive Hours—Other Job Orders				
39	Docking Hours				
40	Actual Overhead Rate				
41	Industrial Overhead Rate Used This Month				
42					
43	Special Appropriation Adjustment Rate—Actual				
44	Special Appropriation Adjustment Rate Used				
45					
46					
47					
48					
49					
50					
51					

(EMERGENCY PRINTING) 992—NNSY—9-12-47—2,000

EXHIBIT 5

ADJUSTED INDUSTRIAL OVERHEAD COMPUTED AND APPLIED IN ACCORDANCE WITH BUS&A DISPATCH 122130 (NOVEMBER 1946)

(1)	Total General Expense (industrial column form 419)			750115.19
(2)	Less Depreciation Including Depreciation in Power Expense Distributed (industrial portion, form 419)			94274.94
(3)	Net General Expense to be Transferred to Shops			655840.25
(4)	Productive Shops and Drydocks Expense (form 419)		565983.99	
(5)	Add: Leave accrued (per BuShips-BUS&A ltr 25 Nov. 1946)		163420.67	
(6)	Total		729404.66	
(7)	Less: Shop and Drydock Depreciation, Including Depreciation in Power transferred (form 419)	72324.36		
(8)	Shop: Sick and Annual Leave Paid (from form 299)	229101.46	301425.82	
(9)	Shop and Drydock Adjusted Overhead			427978.84
(10)	Total Adjusted Overhead Expense			1083819.09
(11)	Total Industrial Overhead Expense Applied			1030175.93
(12)	Under Absorbed Industrial Overhead Expense for Month			53643.16
(13)	Over Absorbed Industrial Overhead, Fiscal Year to Date			217449.01

COMPUTATION OF COMPOSITE OVERHEAD RATE INCLUDING DRYDOCKS

(14)	Total Shop Productive Labor	1252169.28
(15)	Statistical Overhead Rate Before Depreciation, Disability and Leave Adjustments $1324864.36 \div 1252169.28 = 106\%$	
(16)	Adjusted Overhead Rate (less depreciation, disability and shop leave paid, plus leave accrued) $1083819.09 \div 1252169.28 = 86.6\%$	

EXHIBIT 6

has value. Hence, such material when used, either for productive work or as overhead material, has its value carried for cost purposes, although no appropriation charge results. On Exhibit 8, you will see a certain part of the direct material cost identified as an "APA" cost.

Actually, in a sense, Naval Supply Account (N.S.A.) materials are also APA materials in that they have been purchased in the past and paid for by the Naval Stock Fund. The Naval Stock Fund, however, is a revolving Fund, holding the value of the material bought by that Fund in a sort of suspense account in inventory. Then when materials are drawn from N.S.A. stores the using appropriation is debited and the Naval Stock Fund credited with the value of the material drawn.

It will also be noted that on Exhibits 7 and 8 the estimate covers only expected costs of direct labor and direct material. Under certain circumstances; for example, when an estimate is quoted to another government agency or for new construction work, the estimate will also include expected costs of APA material and overhead. For normal operation within the Navy, these items are omitted in the interest of simplicity.

On some of the exhibits, you will note a line labeled "Process Shops." I must give you a brief idea of a "Process Shop." Certain activities such

5ND—NNSY 1 (Rev. 4-48)

JOB ORDER DB

Date: 9–15–49

Job Order Number: 782394401

Appropriation:	Subhead or Treas. Obj.	Project or Allot:	Expenditure Account No.:	Ship or Other Account No.:	Lrl	Brtl	Brstl	Ecl
01601		110415–01	14161	52162				

Ship or Object:

Bureau: Ships (M)

Work Request No.:

Group Name: Propellers

Group No.: S-44

Authority:

Tel. Conv. Comdr. BuShips & Comdr. 9/14/49

Distribution Code	SPECIFICATIONS
	Type of Availability: Restricted (Act. Fleet)
38–0 72 26 41 11 31 23 242 212 228 259 308 338 339 343 345 600 232 Ship	Ref: ComDesLant Dispatch 121913. Allot. MBS 10415/50 TELETYPED 9–15–49 Propellers (2) Service, Port & Stb'd. R.H. & L.H., Replacement of:— Shop 38:—Replace the present port & stb'd propellers with the following propellers: (Stub Propellers from store, APA). Port (4) blade, stock No. S60-P-9116-610 12′0″ diam. L.H., 12′-10″ Pitch, Ser. No. 22474 suitable for 857-890. Stb'd (4) blade, stock No. S60-P-9116-611, 12′0″ diam. R.H., 12′-10″ Pitch, ser No. 22457, suitable for 857-890. Check propeller Fit, also check (dial in place) propeller (Tail) shaft for true (truth). Test upon completion of work. L-350 M-20 T-370 Shop 72:—Service. L-250 M-15 T-265 Shop 26:—Service. L-50 M-10 T-60 Shop 41:—Remove & reinstall interference (rope guards & fairwaters). L-50 M-10 T-60 Shop 11:—Renew rudder zincs. L-25 M- 5 T-30 Shop 31:—Make & Machine parts. (Bolts, nuts, studs, etc.) L-25 M- 5 T-30 Shop 23:—Forge parts. (Bolts, etc.) L-15 M- 5 T-20 Code 242:—Note & furnish correct markings for propellers. NOTE (1): Disposition of propellers removed to be furnished upon docking of vessel.

Estimate No.:	Estimator:	Checked by:	Funds Control:
07037	Winn, W. J.	Barnes	

Labor:	Overhead:	Material:	Total:	Total Mandays:	Days Req'd:
765		70	835	82	3

Job Order Approved:	Work Completed:	Work Satisfactory:
R. L. Pruitt	Ship Superintendent.	Ship Inspector.

EXHIBIT 7

SHOP	MAN DAYS	ESTIMATED CHARGES					
		LABOR	OVERHEAD	Material			TOTAL
				NSA	SMA	APA	
			ESTIMATED CHARGES				
38		350		20			370
72		250		15			265
26		50		10			60
41		50		10			60
11		25		5			30
31		25		5			30
23		15		5			20
Total		765		70			835
			ACTUAL CHARGES				
38		321.28	240.96			13,553.00	321.28
72		263.04	184.12				263.04
26		39.63	29.73	8.40			48.03
41		46.40	34.80	4.00			50.40
11		26.08	19.56				26.08
31							
23							
Total Lab. & NSA Mt'l		696.43		12.40			708.83
Total Overhead			509.17				509.17
Total APA Mt'l						13,553.00	13,553.00
Grand Total							14,771.00
TOTAL							

STATUS OF JOB

COMPLETED:	CLOSED	CANCELLED:
INSPECTED:	COST REPORTED:	FILE BOX:

ADDITIONAL INFORMATION

EXHIBIT 8 NNSY—5-17-48—100,000

as the Gas Manufacturing Plant, the Galvanizing Plant, the Foundries, and Paint Manufacturing Activities are classified as "Process Shops."

The accounting principles used in process shops are quite simple. All expenses in process shops for a specified period are identified as to general character as follows:

Direct material used.
Direct labor used.
Overhead expenses in the shops (excluding repairs to buildings still paid by Yards and Docks)

and are totaled to become the total cost of operating the shop over the specified period. The total units of output over the same period is likewise determined; e.g., for the Gas Manufacturing Shop—the total cubic feet of oxygen produced. Direct division of total cost by total output then gives the cost, or price, of unit output.

The output is then sold as material, by material stub to a productive job order, or by invoice to Naval Stock Fund inventory. Process shops are thus financially self-sustaining in operation, the total returns from sales paying costs of operations.

Now these principles of accounting in process shops seem simple, and should be readily understood. The applications of these principles in practice, however, are very complex, and are quite difficult to follow.

For example, let us discuss briefly the accounting in a foundry.

We first encounter the fact that the output is not homogenous. Various castings are made; iron, steel, brass, bronze and other metals. Each has to be priced individually, and obviously will be of different price. For each of the metals, the direct basic material, brass ingot cost, etc., can be readily identified. The direct labor of melting can be determined. In a like manner, the direct labor and direct material for the mold can be reasonably identified. But much of the foundry labor and material used in melting and molding cannot be identified, reasonably correctly, to a specific casting material, iron, brass, or steel. Hence such cost must be prorated to the various casting types by administrative judgment, such judgment being made on basis of records and past experience. By like means, the expenses of the central foundry offices, and miscellaneous shop expense must be prorated to the various casting types. Finally, it will be realized that the actual price, so obtained, is only available after the fact. Hence the castings must be sold on an estimated unit price basis, and the total return from sales balanced against costs, to determine the under or over-absorbed (profit or loss) for each specific type of casting, brass, iron, steel, etc. This over or under-absorbed balance by month, and to date, of course has a predominating influence on the estimated price of future sales. In total, the procedure becomes quite involved in actual practice, and process shop accounting, as a result, becomes a subject requiring individual study and skill.

Power plant accounting procedures are somewhat similar to those of process shops. But here again the output is not homogenous. Various types of power must be priced separately, such as steam, electric, fresh and salt water, etc. Where practical, direct costs are determined to a specific power. If not practical, costs are prorated to various powers. Ultimately, each power is priced and distributed on a unit price basis. Such prices must again be an estimated price, with an over and under absorbed (profit and loss) balance being maintained for each power with adjustment of the estimated price being made from time to time.

A word about who pays the costs of power. Power plants are not self-sustaining in operation like Process Shops. The entire cost of generating and distributing all types of power is initially borne by "Maintenance, Yards and Docks" appropriations. When the dis-

tribution or sale is to certain types of consumers, such as

Ships (except vessels of Reserve Fleet) at Shipyard using utilities for berthing only, and/or purposes other than repairs or alterations.
Naval hospitals
Naval reserve training centers
Ammunition depots
Marine barracks
Receiving stations and barracks
Coast Guard stations
Navy-sponsored activities
Other government departments
Process shops

an appropriation adjustment, credit to "Maintenance Y&D," debit to the appropriate appropriation, is made. The remainder of power costs are considered Industrial (or Non-Industrial) and remain as overhead and a charge to "Maintenance, Yards and Docks," appropriations.

I have tried to show what are industrial costs, how they are made up, and who (Bureau or appropriation) pays for certain parts of these costs. So far, I have not tried to show how such portions of cost are paid, i.e., the procedural methods. The following should go far to show methods of paying overhead costs:

(a) The direct labor and direct material used in productive work is paid for directly by the Bureau and appropriation ordering or requesting such work. As previously stated, such order or request may come direct from the Bureau or Office, or it may come from other Command channels, such as the Commander in Chief of the Atlantic Fleet (or his type Commanders). Such Commands, however, have previously received from the appropriate Bureau concerned an authorization to expend that Bureau's funds, for certain specified purposes, up to, but not exceeding, certain amounts. In essence, therefore, they operate simply as agents of the Bureaus concerned.

(b) Overhead costs are paid for mostly by "Maintenance, Bureau of Ships," "Maintenance, Yards and Docks," and "Maintenance, Supplies and Accounts." When occurring, they are charged direct to allotments issued from those Bureaus, to the Shipyard Commander, for the purpose of payment of those costs. These allotments are generally known as the Shipyard Commander's Budget. Some years ago, some portions of the overhead costs were actually charged to the customer; i.e., the Command, ordering or requesting the work. No matter how paid, it should be realized that the magnitude of the industrial cost is not changed. Thus the pre-war procedure was but another procedure for paying overhead costs. Sometimes, the pre-war method did have the effect of shifting the burden of part of the overhead costs to other Bureaus or to other appropriations. For example, presently if the Bureau of Ordnance orders work done in Naval Shipyards, for all practical purposes, it pays no portion of the overhead costs, such cost being borne by the Bureaus of Ships, Yards and Docks, and Supplies and Accounts. In pre-war years, when some of the overhead cost was charged to the customer, the Bureau of Ordnance would have paid part of the overhead cost.

If at this time I have succeeded in giving a fair understanding of industrial costs, how they are made up, who pays

them, and how they are paid, I shall feel gratified.

Now, I do not wish to confuse the subject, but there are several important variations in the system which I should cover briefly.

Naval Shipyards often do work for other government activities, and sometimes they do work for private parties. They also may do a considerable amount of New Construction and Conversion work for the Navy. When work of the foregoing categories are accomplished, the industrial overhead is costed exactly as for regular Navy work. However, in these instances, some, if not all, of the industrial overhead is actually charged the customer.

When Shipyards do work for other government activities, the full industrial overhead rate, not including Depreciation or Disability, is applied to the labor charges, and the full industrial cost thus obtained is charged the other government activity. But, of the funds so obtained, only a portion of the overhead costs, perhaps 50%, rebounds to the credit of the Bureau of Ships maintenance appropriations, the remaining 50% of the overhead cost being turned in to the Treasury as "Miscellaneous Receipts." This distribution may seem peculiar, as perhaps it is. I make no attempt to explain the logic of such a distribution.

When work is done for private parties, the same costing and charging procedure is followed but all overhead charges are turned in to "Miscellaneous Receipts, Treasury." None of the charge for overhead rebounds to the credit of the Bureau of Ships appropriations.

When Naval work of a New Construction or Conversion character is undertaken by a Naval Shipyard that work pays a great portion, if not all, of the overhead cost of performing the work and the amount of such overhead paid is credited to Bureau of Ships Maintenance Appropriations. The technical means of making such charges to the Shipbuilding Appropriation, and credits to Maintenance Appropriations will not be covered in this paper.

No doubt, some readers of this article may detect rather flagrant technical omissions or errors. To excuse myself, if such are detected, I ask the readers to remember that this article is written, not from the background of an accounting officer or an accounting expert, but from the background of a specialist officer (ED Classification) who has endeavored over the past years to understand and use the accounting system as an assistance in management.

It may be appropriate at this time to introduce some personal opinions and ideas of my own on the general question of Cost Accounting.

One question constantly arises: Does the Navy use a true industrial overhead cost? The answer to this can only be that it does not. I have already shown that depreciation and disability, although calculated, are dropped in the final determination of industrial costs. There can be no doubt that such costs are real, and must be used continuously in commercial practice to avoid bankruptcy. Other overhead costs encountered in commercial practice, are not found listed in the shipyard industrial costs. Such items are sales promotion, cost of sales, taxes, insurance and, of course, profit whether or not profit be called overhead.

On the other hand I do not wish to leave an impression that the Naval Shipyards have a great advantage over commercial industries in the matter of overhead burden. Naval Shipyards must absorb certain overhead costs not found in commercial industries. Many commercial industries have liberal vacation and sick leave policies, yet, to my knowledge, none have policies as liberal as the Government grants its employees, including those in Naval Shipyards. The difference is significant and a very considerable portion of the overhead burden in Naval Shipyards can be attributed directly to annual leave, sick leave and

holiday pay. Also, Naval Shipyards carry a unique overhead burden caused by what I might call the industrial ills of high and extremes of plant capacity. A commercial plant, at least to some extent, can pick and choose its work. It can manufacture the articles it selects. It can accept repair work or manufacturing orders compatible with a limited plant capacity. Naval Shipyards, on the other hand, for military reasons if for no other, must have a plant capable of accomplishing for Naval vessels, repair work of any degree of complexity and of any magnitude and size. This requires that the plant facilities of the Naval Shipyards must be very highly developed. Besides finding many single purpose machine tools of extremely high precision, we also find, machine tool equipment of extremely high capacity. Lathe equipment ranges from the finest of jewelers lathes to lathes capable of swinging items twelve feet in diameter or turning and boring shafting eighty feet long. Twenty or twenty-two foot boring mills are not uncommon. I know of at least one thirty-foot mill. Crane capacities range up to four hundred tons. Dry docks over a thousand feet long with forty feet of water, or more, over the sill are available in the larger Yards. Obviously much of this high capacity or specialized equipment can be fully utilized but occasionally and, while the Navy absorbs no depreciation charge in its overhead rate, there is a large maintenance and upkeep cost on these tools, all of which adds to the overhead burden.

In the exhibits a rate of about 85% is shown. The question often arises—what is an appropriate or satisfactory overhead rate? Is this rate of 85% high or low? In answering this question, it must first be accepted that the quantitative value of the overhead rate, or the actual amount of the overhead cost, is not an item which determines or measures whether a plant is operating efficiently or not. A plant with low overhead may be, and often is, considerably less efficient than one with high overhead, in that the total unit cost of the work output item is considerably higher. A simple example will illustrate this point. Suppose we have a blacksmith forging an item of equipment. The essential plant property he will use is a small forge, an anvil and a few hammers and other hand forging tools. At any rate, the plant investment would be quite low, and the maintenance charge correspondingly low. He may work on each forging an entire day, and an entire day's labor would thus be charged to each individual forging. However, the overhead rate would be relatively low—perhaps 20%, and consequently we would have a case of a unit of output to which high direct labor costs are charged, but relatively low overhead costs are charged; that is, a low overhead rate obtains. Now suppose that we manufacture this forging by means of a drop forging process. We must have a drop forge which in itself is an expensive item carrying a large maintenance charge and which requires a rather large power load, with its cost. There is also the expense of making dies, and replacing dies when worn. Dies are a very expensive item, and the cost of maintenance on such dies will be considerable. But, the operation now is a very rapid operation. Forgings which perhaps previously took an entire day's time to produce are fully produced and completed in but one operation of the drop forging press. It may now take two or three men to operate the drop forging press whereas before we had but one man, the blacksmith, at work. Moreover, with the units of output being made so rapidly, it is no longer possible to identify the work of any one of these men directly to a single unit of output. And as a consequence, we have as the result the production of an item with no direct labor charges and all wages, power, depreciation, etc., being charged as overhead. In which instance, the

overhead rate would be infinity. Certainly, however, we must admit that the latter operation is far more efficient, and the unit of output would be far cheaper than when the forging is produced by hand methods.

Another example: an under-development of service activities usually means that total cost of work will be excessive due to high labor costs, although overhead costs may be low. It is nearly an axiom that if supply functions, crane and transportation services, etc., operate at their maximum efficiency, then production functions will operate at lowered efficiency. The service activities named must be manned to meet the averages of normal *peak* demands of the productive forces, which usually means that those service activities are operating at less than maximum efficiency a large part of the time. It requires a nice judgment in the managerial group to determine the level at which service activities should operate to obtain the best practical operating efficiency of the production force, and the *least total cost* of output.

On the other hand, I wish to warn that a high overhead rate is not at all presumptive of efficiency. It may be that we have simply overburdened the productive process with far too many, or too highly developed, service activities or overhead functions.

It should be realized, too, that the quantitative value of overhead is, to some extent, dependent on rules of accounting, themselves subject to change. To that extent "overhead" is "what we make it." By the rules, all drafting, design and transportation costs are classified as overhead. Yet it will be easily recognized that, in many instances, the work of draftsmen, engineers, and truck drivers could be identified to a specific productive job order. If such men, so identified, were to be charged directly to the productive job order, overhead costs and overhead rates obviously would be reduced.

If we can then agree that an overhead rate, in itself, is not indicative of efficiency, what is indicative of efficiency in the overhead rate? The most significant indication is *changes* in that rate, or changes in the amount of overhead being charged. Such changes must come about for some reason. The detailed constitution of overhead costs, and the changes, are the items to which the efficient manager of an institution focuses his attention, not to the quantitative value. From the above, I think we will all agree then that there is no quantitative figure that represents a fair or a proper value of the overhead rate.

Currently, a question popular in Naval circles—is the present method of paying overhead costs, that is, by direct allotment to the local level, as satisfactory as a method requiring the recipient of the productive work to shoulder his equitable share of the overhead? There are two sides to this question. Either method has advantages and disadvantages. It has sometimes been found that it is quite difficult to balance the overhead funds required in a Naval Shipyard with the productive funds that may be made available for work. It is possible for the Bureau of Ordnance to have ample funds to accomplish certain manufacturing work. Yet the Bureau of Ships or the Bureau of Yards and Docks, or the Bureau of Supplies and Accounts, may be so short of overhead funds that there will not be sufficient funds in the Commander's Budget to pay the leave, supervision and miscellaneous costs of overhead needed to support the productive workers required to perform the work of the Bureau of Ordnance. This is a real problem and is constantly encountered by the Bureaus and the Shipyards. It requires a great deal of administrative and management talent to keep the funds balanced, and the results are not always completely satisfactory, but there is usually a solution to this problem by one means or another.

To one experienced in both methods of paying for overhead, there is no question that the present method, where overhead is paid by an allotment to the Shipyard Commander, has the effect of forcing a great deal more attention of the Yard management on elements of overhead. In the prewar days when productive work itself carried its share of the overhead, or a sizable portion thereof, there was a tendency not to worry too much about what the overhead costs were, since the customer was going to pay anyway, and since he had no choice but to pay—his ability to shop around between Naval Shipyards was extremely limited, and except in rare instances, he couldn't turn to commercial plants,—so why worry. Now, however, with the overhead costs payable by allotment of which the Commander himself has the cognizance, costs of the overhead type are very carefully considered and scrutinized continually. There can be no doubt that such scrutinization tends toward efficiency of operation, and reduction of actual overhead to the required minimum. A disadvantage of the present system of paying overhead costs is that the customer fails to obtain an appreciation of the *true* cost of work, or what he gets. Since he is paying but the direct labor and direct material costs, he is often prone to believe that the work is not very expensive, particularly when compared with other commercial institutions, who make a charge to him for overhead, profit, etc. This is but an example of mental confusion between appropriation charges and industrial costs of work. The pre-war method served a better purpose in convincing the customer of the real cost of work, even if, at that time, he usually thought it was too high.

Another question constantly to the front in present days: Why should not Maintenance, Bureau of Ships, pay all the overhead costs? The reasoning back of this question is—the Bureau of Ships has management control of Naval Shipyards, and therefore, should be financially responsible for all overhead expenses within those yards. Certainly such procedure is possible and could be arranged. To do so, would, however, require an adjustment in appropriations between the three Bureaus now paying the overhead expenses with Maintenance Bureau of Ships being increased, and the appropriations of the other two Bureaus decreased. Thus changing the present method of paying overhead costs would involve high policy decisions even outside the Navy Department. Such a change is not practical until the appropriation adjustments are made and become effective. This would require different budgeting in presenting appropriation requests to Congress, accomplishment of which would require full concurrence of the Bureau of the Budget and of Congress. To date such full concurrence has not been forthcoming, but I understand a change toward the above ends will be made in the next fiscal year.

Is the present accounting system satisfactory? In answering this question, the Navy's Accounting System must be considered not only in the light of its service to the Navy but in light of the service given to Congress. It is to Congress that the Constitution gives the power to raise revenue and make appropriations. Congress has expressed its requirements and needs in meeting this task, in many laws and rules relating to estimates, appropriations and costs. By and large the present accounting system has seemed to satisfy most of the Congressional requirements. In other words, the present system serves satisfactorily to render service to Congress. For the Navy service, it is more difficult to make the same statement. Accounting is complex and it does not have wide understanding. As a consequence, much confusion exists in the minds of officers and civilians of the

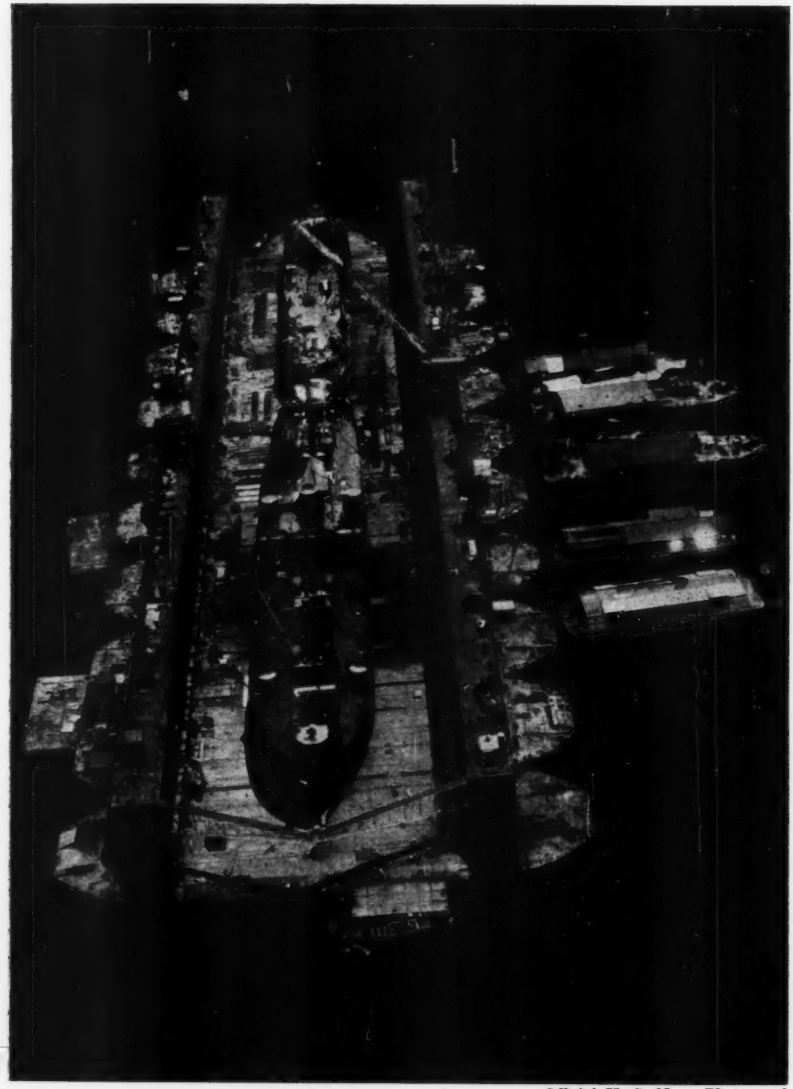
Navy concerning certain of the accounting practices. A major confusion arises by failure to make proper distinction between appropriation charges and industrial costs. Lack of knowledge of how certain industrial costs are paid contribute strongly to confusion. A lack of realization that accounting is but a means of administration and is not the end in itself also contributes to the general confusion. Many general complaints are lodged against certain accounting rules, such complaints apparently being registered without the realization that the rules are subject to change if need be. If we can but remember that accounting is a tool of administration, a tool for proper management, and does not control management, then we perhaps will have a clearer approach to understanding. Accounting should not dictate what is to be done, it should simply keep records of what has been done so that the managers of the Navy can understand what is going on. In my opinion, much of the present dissatisfaction of the accounting system among Naval personnel can be attributed to the failure of Naval personnel to thoroughly understand accounting principles, and lacking that understanding, their inability to use the accounting system as a proper tool to accomplish desired ends.

The Navy does receive one dividend from the accounting practices in naval shipyards that is rarely appreciated outside those officials intimately connected with the management of naval shipyards. The present accounting system, with modifications, has been in effect for about 40 years. This span of time, with but few exceptions, covers the working life of nearly all employees and officers connected with shipyard management and operation. Among responsible civilians and officers, there is a full realization that employment; i.e., stability of employment, in a naval shipyard is dependent entirely on work assigned. Assignment of work brings with it funds, and these funds pay the wages of employees. Work assignments to a naval shipyard are somewhat similar to "orders booked" by a commercial firm. The continued life of an organization depends on "orders booked," and for a naval shipyard depends on work assigned. As a result, there is a sustained incentive to do good work as cheaply as possible, and to give the customer satisfaction. The consumer, it may be true, is internal within the Navy, yet it is fully realized by a Shipyard that continued work assignments rest on a firm foundation of good work records of the past. This feeling and desire for work and more work permeates the entire organization, organized employees groups as well as the management groups. There can be no doubt that this appreciation of the requirement for more work tends towards improvement in operation and efficiency at all times.

The government in recent years has initiated a rather large program, headed up by the Bureau of the Budget called "Work Measurement." The Navy is pursuing its part of that program. Considerable thought and effort has been applied to the mechanics and operation of the program. The thought back of the program is to develop standards for work, standards of manpower requirements, perhaps standard material requirements, and other items. Then with these standards in mind, actual measurement of work output, with subsequent verification or back check against the original standards being made. If one stops to think it should be realized that, in our industrial cost accounting system, we have a fully developed and operating work measurement program. Productive work is specified in great detail, estimates are made, and the final costs are measured against those estimates. The estimates are made against developed standards of accomplishment or by detailed process analysis. What is this but a full-fledged work measurement program? It is true that because the work

output of a Naval Shipyard is extremely varied, the work on no two ships being identical, nor the work on any two items being identical, the ability to develop constant standards of work is unattainable. It is impossible to say that a pump should take so many dollars or hours of work. The pump may be of different size than another pump or, it may be, if of the same size, in different condition, and the one pump may require a great deal more repair work than the other. Nevertheless, even with these different types of output work, each instance is specifically estimated, the work accomplished, and the resulting cost in time and money measured against the estimate.

My understanding of cost accounting in the Navy has come to me over the many years as a result of experience, sometimes difficult, augmented by constant reference to the Bureau of Supplies and Accounts Manual where the present system is technically outlined in detail. For those individuals that wish to pursue further the study of Naval Shipyard Cost Accounting, as regards the technical details, and to fill in the parts of this article poorly covered or omitted, the most important references are the Bureau of Supplies and Accounts Manual, Volumes VI and VII. In those volumes, Chapters 4 and 7 of Volume VI and Chapters 2 and 5 of Volume VII are the most pertinent chapters.



Official U. S. Navy Photograph

Floating Dry Docks—"Field Hospitals" For Navy Ships

THE DEVELOPMENT OF GRAPHIC AIDS TO PRELIMINARY DESIGN

CEDRIC RIDGELY-NEVITT

THE AUTHOR

CEDRIC RIDGELY-NEVITT, upon graduation from Webb Institute of Naval Architecture in 1939, was awarded a Society of Naval Architect's scholarship for advanced work at Massachusetts Institute of Technology. He received an S.M. in Marine Engineering in 1940. For the past five years, he has been Assistant Naval Architect for Kindlund and Drake, in charge of preliminary and detail design work. Prior to this, he was head of the Hull Technical Division at Cramp Shipbuilding Company. He has also served as a part-time professor of Engineering at Webb Institute of Naval Architecture.

INTRODUCTION

The time required for a naval architect to produce a satisfactory preliminary design for a new ship is a function of the amount of data in his possession for similar or related vessels, together with the degree of divergence in characteristics from the most nearly similar ship whose particulars are known. In the following analysis, the author has investigated certain features which can be delineated graphically; i.e., items which show a definite trend over a wide range of vessels.

Graphical presentation has several distinct advantages to the designer.

1. A given value can be checked at a glance.
2. Any new data can be added on existing curves and, as later trends are established, new curves can be drawn accordingly.
3. Numerical errors resulting from the use of formulae are reduced to a minimum.
4. Variations from normal or average practice may be discerned, as soon as they appear, and the reasons for such variation investigated before subsequent work is undertaken.

The author has, therefore, not attempted to compile a complete treatise on preliminary design, but, rather, has concentrated on general, related characteristics that may be useful for normal types of ships with large changes in important variables. He has, in all cases, presented the reasoning behind and development of each individual plot, so that another designer, who may specialize in a particular type of vessel, can prepare similar curves, covering a more limited range, but adapted to his specific requirements.

DISPLACEMENT

Before a definite preliminary design of a normal freighter or tanker can be undertaken, the speed and deadweight carrying capacity must be stipulated or estimated. The term, speed, is assumed, in all subsequent analyses, to mean the required service speed at sea. Then as a first step in the design of the ship, the naval architect must estimate the loaded displacement required.

Figure 1 shows values of deadweight divided by displacement for 27 large and medium-sized American cargo ships built within the last fifteen years. These deadweight-displacement ratios are plotted as ordinates with corresponding speed-length ratios as abscissae. These items are defined as:

$$\text{Deadweight-Displacement Ratio} = \frac{\text{D.W.}}{\Delta}$$

$$\text{Speed-Length Ratio} = \frac{V}{\sqrt{L}}$$

D.W. = deadweight at designed load draft in long tons.

Δ = displacement at designed load draft in long tons.

V = service speed in knots.

L = length between perpendiculars in feet.

Although, from a resistance standpoint, it would be more logical to use the load water line length, the length between perpendiculars is usually more convenient and more representative of the actual size of the ship. The former is a function of end profiles, which will not be established until a later stage in the design. In addition, when published data are referred to, the latter value may be the only one available. Hereafter, length should be interpreted as that between perpendiculars unless it is specified to the contrary.

Figure 1 has introduced length as a variable, even though this has not, as yet, been determined. It is not too important a one, at the moment, and can be approximated from a known vessel of comparable deadweight. Since a range of ± 0.03 from mean values of deadweight-displacement ratio is shown,

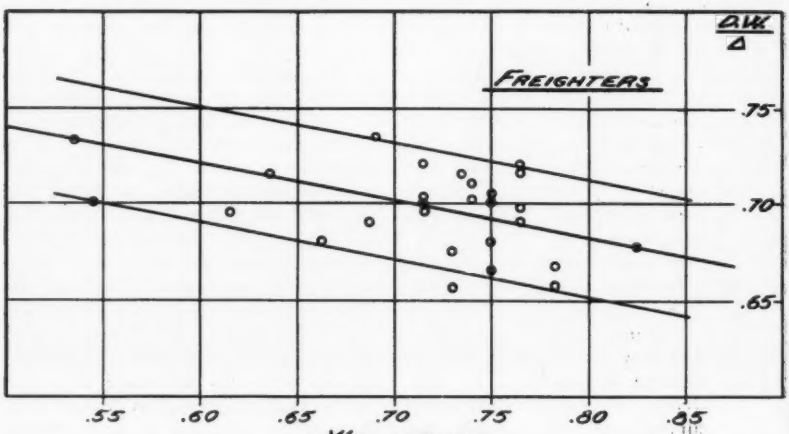


Fig. 1.—Deadweight-Displacement Ratios for Freighters.

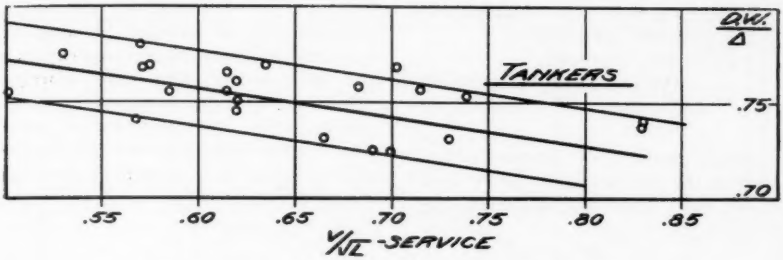


Fig. 2.—Deadweight-Displacement Ratios for Tankers.

a certain amount of judgment is required to select a suitable value. The following items contribute toward lower values of the ratio.

1. Riveted construction.
2. Strengthening for navigation in ice-filled waters.
3. Extensive cargo-handling facilities.
4. The refrigeration of certain holds; in fact an all refrigerated ship will fall below the lowest line in the figure.
5. Limitation in draft for shallow water service or from a flooding standpoint.
6. Large numbers of holds or 'tween decks.
7. The use of direct-connected diesel machinery.

Those contributing to higher values are:

1. Welded construction.
2. Extensive use of light alloys for superstructure and fittings.
3. Elimination of cargo handling in bulk-cargo vessels.
4. Full-scantling ships taking advantage of the minimum statutory freeboard.

Figure 2 shows similar values and trends for 23 large and medium size American tankers of normal type. Because there are fewer variations between individual ships, a range of ± 0.02 is shown.

Features contributing to lower deadweight-displacement ratios are:

1. Riveted construction.
2. Flat plate bulkheads.
3. The use of scantlings greater than classification requirements as an extra corrosion allowance.
4. Provision for split cargoes with extra tanks, cofferdams, piping, and pump rooms.

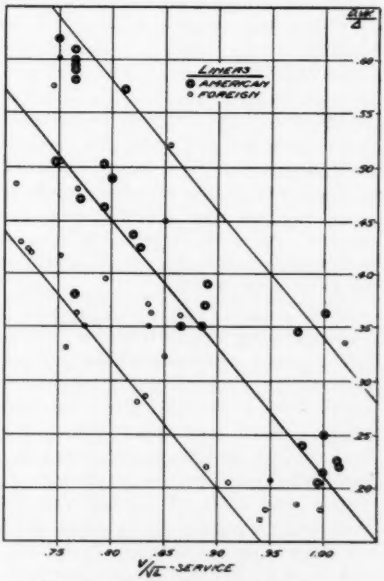


Fig. 3.—Deadweight-Displacement Ratios for Liners.

Higher values may be obtained with:

1. Welded construction.
2. Fluted bulkheads.
3. Large tanks and limited pumps and piping for a single grade of cargo.

Large tankers of unusual design, such as the *Phoenix*,[1] may show higher values than those covered in Figure 2.

Figure 3 is not intended for design purposes, but is included to complete the general picture of carrying capacity for the passenger ship case. Foreign vessels are included to widen the range which varies from 0.60 for a cargo liner with a limited number of passengers to 0.20 for a superliner or cross channel steamer.

LENGTH

Having completed a first estimate of load displacement, by using the simple ratios plotted in Figure 1 for a freighter; Figure 2 for a tanker; or by an extensive analysis, not detailed here, for a passenger ship, the next step is to establish a trial length for the vessel being designed. This is a difficult one, for a designer can find little published information to aid him in arriving at a satisfactory answer. Certain aspects of the problem have, however, been considered by Messrs. Comstock[2,3] Baker[4] and Lachowski.[5] The first was formulated some years ago and needs modification, the second represents British practice for a specific type of vessel, and the third, based on work by Hecksher of the Hamburg tank, is based on a length for minimum resistance. As a result, none is a completely satisfactory means of attacking the particular problem.

Every designer has solved the length problem, to his own satisfaction; therefore, a group of previous solutions are available as a logical guide for future work. For a starting point, in this analysis, beam and draft values for 37 American designs were plotted opposite length in Figure 4. Shallow draft vessels, and craft of unusual proportions were excluded. The ships used are ocean and coastwise freighters, tankers, and liners; all are merchant

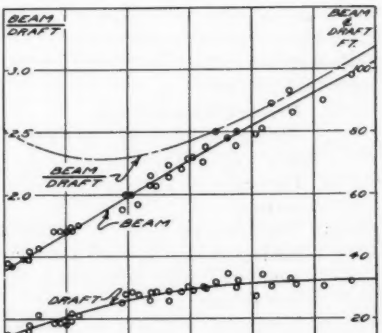


Fig. 4.—American Ship Proportions.

ships with relatively full midship sections. Mean curves of beam, draft, and the beam to draft ratio were drawn. This establishes a set of average ship proportions with length as a basic variable, but does not include the fineness of the hull.

Figure 5 shows prismatic coefficients for almost 50 ships as a function of speed-length ratio. The mean curve is identical with Baker's 1943 proposal.[4] If it is shifted, horizontally, 0.10 speed-length ratio in either direction, the resulting band will include all the vessels

[1] References are given in the bibliography.

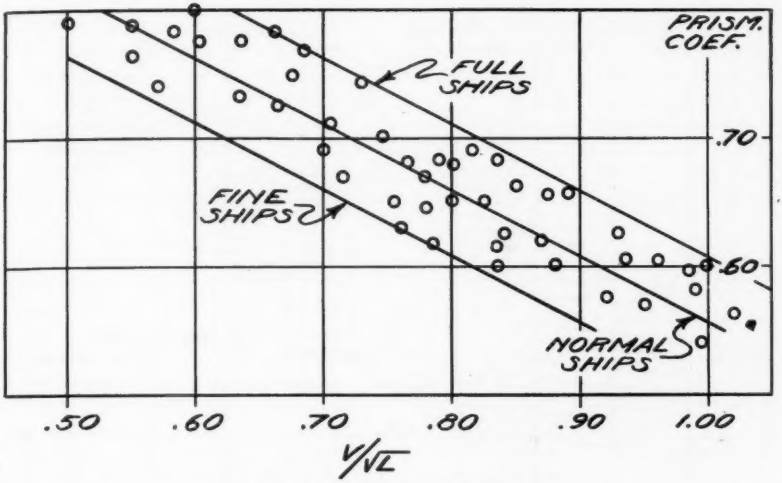


Fig. 5.—Prismatic Coefficients.

shown. The top line represents full ships, the mean line normal ones, and the bottom line fine vessels. If attention is limited to the ordinary merchant hull form with little or no dead rise, no greater error will be introduced by using a midship section coefficient of 0.97, in order to convert prismatics to block coefficients. Then, Figures 4 and 5, together with the foregoing assumption, define three series of ships with length between perpendiculars and service speed as primary variables. Beam, draft, prismatic coefficients (and hence blocks) can be read off the curves; displacements, displacement-length ratios, and the beam-draft ratio can be computed. Furthermore, these elements are all the ones required for estimating wetted surface and effective horsepower by means of Taylor's Standard Series.[6]

The author has made these computations for vessels from 200 to 800 feet in length with displacements of 2000 to 40,000 tons in salt water. Figure 6 shows the displacement-length ratios derived, and Figures 7, 8, and 9, respectively, the speeds, lengths, displacements, and effective horsepowers for the full, normal, and fine series of vessels. In the development of these curves length was considered the basic variable. Conversely, once they have been developed, displacement and speed may be used in determining a suitable length.

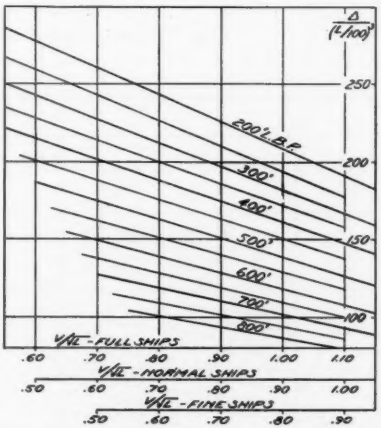


Fig. 6.—Displacement-Length Ratios.

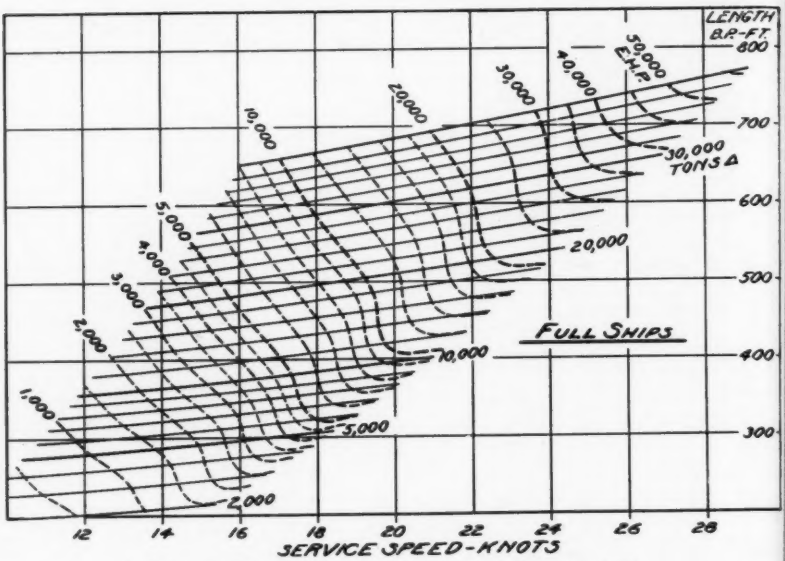


Fig. 7.—Displacement, Length, and Speed—Full Ships.

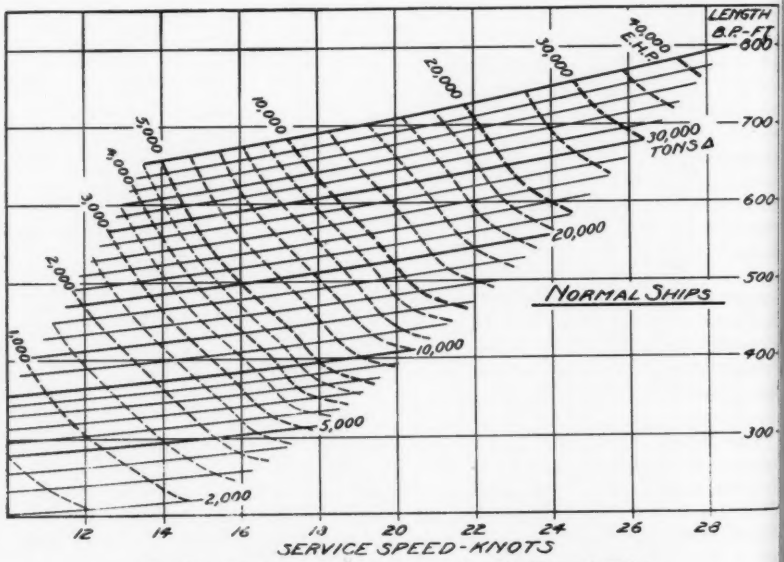


Fig. 8.—Displacement, Length, and Speed—Normal Ships.

Figure 8 may be considered to represent average American practice, as determined in Figures 4 and 5, the only assumption involved being the midship section value of 0.97. From this figure, as soon as displacement and speed are known, a trial length and first approximation of E.H.P. can be read off. Similar values for a fuller and finer ship can be obtained from Figures 7 and 9 and the effect of variation in horsepower with length determined.

Although not too important, at this stage, both displacement and horsepower include the effect of appendages.

The shapes of the horsepower curves are quite interesting. They are smooth and fair for the fine ships. In the normal ones, unfairnesses appear where bow and stern waves interfere. In the full ships, the interference is quite marked; in fact, in some regions, it is possible to increase speed, at constant displacement and a slight increase in length, without an increase in power.

Table 1 compares the values taken from Figures 7, 8 and 9, with data for eight vessels built in this country during the last ten years. It should be noted that the tankers, in particular, conform quite closely to the full ship values of Figure 7. For these ships, in order to have sufficient room for their machinery at the stern, a rather high prismatic is chosen. For a cargo ship or passenger vessel, with machinery amidships, one can take advantage of a finer prismatic and reduced power requirements.

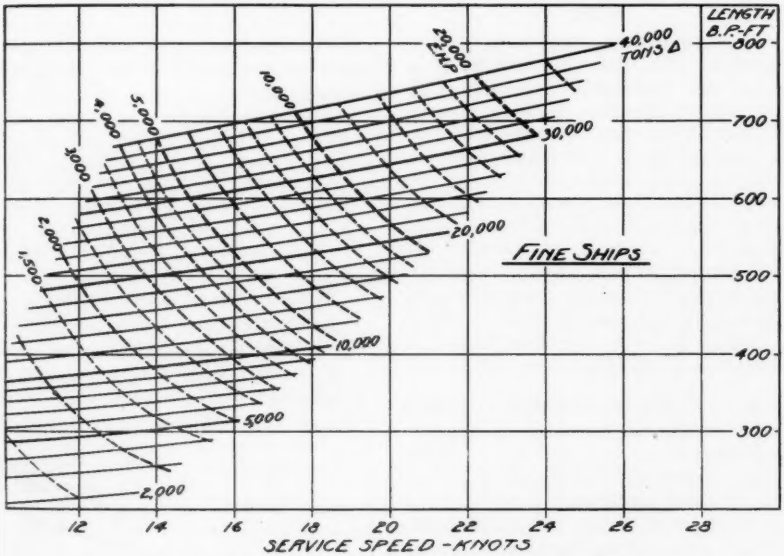


Fig. 9.—Displacement, Length, and Speed—Fine Ships.

TRANSVERSE AND LONGITUDINAL METACENTERS

The previous sections presented curves for use in making a first approximation of displacement and length for a normal ocean or coastwise vessel whose draft was not severely restricted by service conditions.

TABLE I

Type	Displ. Tons	Service Speed	As Built		Figure 7 Full Ship		Figure 8 Normal Ship		Figure 9 Fine Ship	
	S.W.	Knots	E.H.P.	L.B.P.	E.H.P.	L.B.P.	E.H.P.	L.B.P.	E.H.P.	L.B.P.
Twin Screw Atlantic Liner	34,700	22.5	21,500	661	23,000	665	20,500	690	19,000	715
Twin Screw Tanker	23,200	18.0	8,650	525	8,600	525	7,900	542	7,500	560
Single Screw Tanker	21,200	15.0	4,500	488	4,400	490	4,200	505	4,100	520
Twin Screw Pass. and Cargo Liner	14,000	16.5	4,350	472	4,800	425	4,400	440	4,200	455
Single Screw Freighter	14,000	15.5	3,670	445	3,900	420	3,600	432	3,400	445
Single Screw Freighter	12,900	14.0	2,580	395	2,650	400	2,450	410	2,400	420
Single Screw Freighter	7,450	11.0	900	320	900	315	870	325	850	335
Single Screw Tanker	2,200	10.0	—	213	—	205	—	210	—	215

Beam is usually determined, in conjunction with draft and suitable prismatic and midship section coefficients, to provide adequate stability. Let us consider, therefore, the variables required to locate the position of the transverse metacenter. At present, this will be investigated only at the load draft, but later work will widen the field.

The height of center of buoyancy above baseline can be easily, and quite accurately, estimated by:

$$KB = \frac{C_w}{C_b + C_w}(H)$$

KB = distance from baseline to vertical center of buoyancy.
H = draft.
C_w = water plane coefficient.
C_b = block coefficient.

It is assumed that the above coefficients are based on the length between perpendiculars. Figure 10 may be used as a guide for estimating the waterplane coefficient for almost all types of ships except those with transom sterns. The 46 vessels plotted include liners, tankers, freighters, tugs, trawlers, ship-shaped barges, and even a submarine.

The transverse metacentric radius, or height of metacenter above center of buoyancy is:

$$BM_t = \frac{C_i L B^3}{C_b L B H} = \frac{I}{V^1} = \frac{C_i B^2}{C_b H}$$

BM_t = height of metacenter above center of buoyancy.
I = transverse moment of inertia of waterplane.
V^1 = volume of water displaced.
C_i = transverse inertia coefficient.
L = length between perpendiculars.
B = beam at waterline.

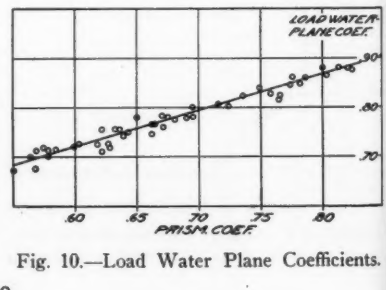


Fig. 10.—Load Water Plane Coefficients.

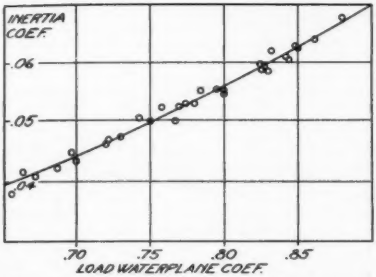


Fig. 11.—Transverse Inertia Coefficients.

Figure 11 shows values for the transverse inertia coefficient and is equally wide in its application as Figure 10. Once again the transom stern case must be excluded. These two figures show that, once the prismatic and midship section coefficients and principal dimensions are fixed, we have all the necessary elements required for determining the location of the transverse metacenter. For purposes of calculation, let us assume, as we have done previously, a midship section coefficient of 0.97 for the normal type of merchant ship. Figure 12 shows ratios of KM to draft for ships with a midship section coefficient of 0.97, prismatics varying from 0.55 to 0.825 and beam to draft ratios from 2.0 to 4.0. By the use of the correction curve, at the bottom of the figure, the results may be modified for midship section coefficients from 0.95 to 0.99. The correction is added for values below 0.97 and subtracted for those above.

Longitudinal stability can be examined in an analogous manner:

$$KM_L = \frac{C_w}{C_b + C_w}(H) + \frac{C_{iL}L^2}{C_b H}$$

KM_L = height of longitudinal metacenter above base line.

C_{iL} = longitudinal intertia coefficient.

Here, however, we assume the center of gravity of the ship is at the load waterline and plot values of GM_L/H, the longitudinal metacentric height divided by draft. Figure 13 gives the values of longitudinal inertia coefficient and

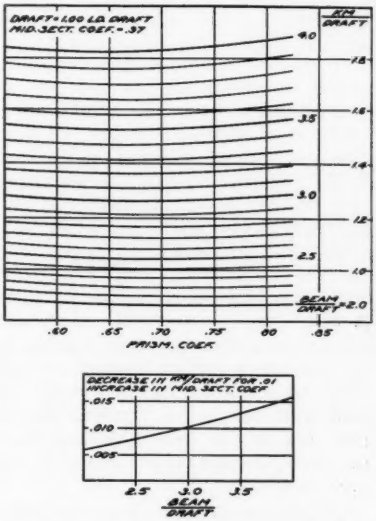


Fig. 12.—Transverse Metacenters at Load Draft.

Figure 14 summarizes the calculations covering length to draft ratios from 10 to 30, prismatics from 0.55 to 0.825, with a midship coefficient of 0.97. Once again, the lower part of the figure gives corrections for midship coefficients within the range of 0.95 to 0.99, but this refinement is usually omitted. Extreme accuracy in longitudinal stability is rarely required at any early stage of the design process.

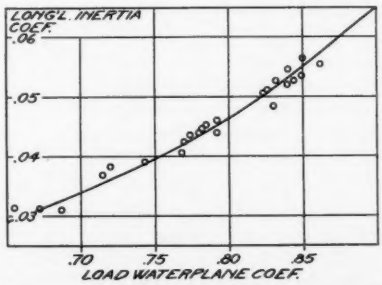


Fig. 13.—Longitudinal Inertia Coefficients

FORM CHARACTERISTICS AT OTHER DRAFTS

In the last section, curves were presented for estimating the position of longitudinal and transverse metacenters from the principal dimensions and form characteristics of a vessel in the load draft condition. It may be necessary, however, to examine variations in displacement and stability with draft for some types of vessels. A tanker will have its minimum metacentric height at load draft with free surface in all tanks. A cargo ship, on the other hand, will usually have the minimum stability at the end of a voyage started at load draft, when all the fuel and water are used from the double bottoms. And a fine ship, of the liner type, will have the minimum metacentric height in the light condition.

The element most useful for representing vertical distribution of displacement is the vertical prismatic coefficient, C_v:

$$C_v = \frac{C_b}{C_w}$$

This has been already introduced, in effect, by using the equation for vertical center of buoyancy:

$$KB = \frac{C_w}{C_b + C_w} (H)$$

This may be transformed to:

$$KB = \frac{1}{C_v + 1} (H)$$

In order to ascertain the ordinary range of vertical prismatics, values of water plane coefficient were taken from Figure 10, and block coefficients obtained by multiplying prismatics by midship coefficients varying from 0.95 to 0.99. Figure 15 shows that, for the average ship, there is a definite relationship between longitudinal and vertical prismatics. Figure 16 gives corresponding approximations of the vertical center of buoyancy, which were, incidentally, used in the construction of Figures 12 and 14.

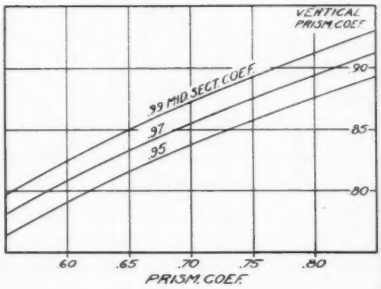


Fig. 15.—Vertical Prismatic Coefficients.

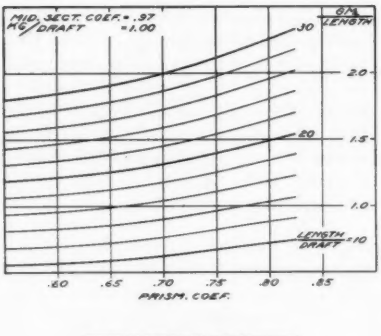


Fig. 14.—Longitudinal Metacentric Heights at Load Draft.

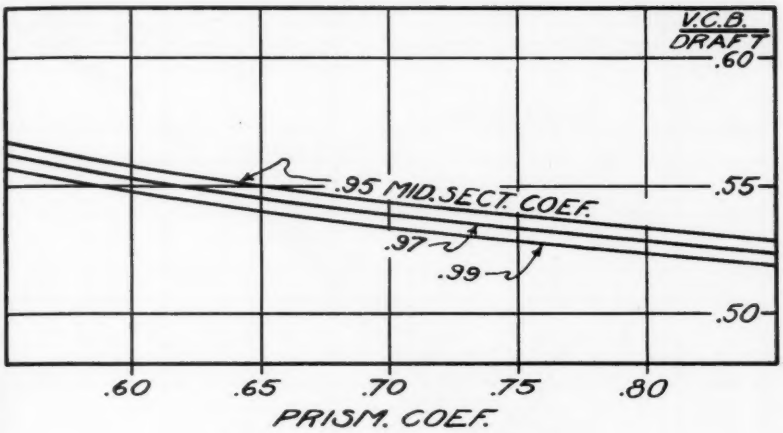


Fig. 16.—Vertical Centers of Buoyancy.

Two excellent investigations of form characteristics with varying draft have been published: one by Comstock[7] and the other by Dyer[8], both of which give quite consistent results:

According to the latter:

$$\frac{\Delta}{\Delta_1} = \left(\frac{H}{H_1}\right)^{\frac{1}{C_v}}$$

$$\frac{C_b}{C_{b1}} = \frac{C_w}{C_{w1}} = \frac{A}{A_1} = \left(\frac{H}{H_1}\right)^{\frac{1-C_v}{C_v}}$$

Δ_1 = displacement at an intermediate draft.
C_{b1} = block coef. at an intermediate draft.
C_{w1} = waterplane coef. at an intermediate draft.
A = waterplane area at load draft.
A_1 = waterplane area at an intermediate draft.
H_1 = intermediate draft.

Plots of these variations are given in Figures 17 and 18; these can be used in conjunction with a vertical prismatic from Figure 15 to obtain displacements, tons per inch, and, hence, deadweights of a vessel once these have been determined for the load draft condition.

The author has used neither the Dyer nor the Comstock formulation for computing metacenters but, for reasons of consistency, has calculated them, exactly as in the previous section, based on the following assumptions:

1. Block coefficient and water plane coefficient variations are taken from Figure 18.
2. Transverse inertia coefficients are taken from Figure 11.
3. Vertical prismatics are in accordance with Figure 15.
4. For determining the vertical centers of buoyancy, there is no variation of vertical prismatic with draft.

Figures 19 through 22 present the results of the foregoing assumptions and calculations for the 0.97 midship section case, each with its correction

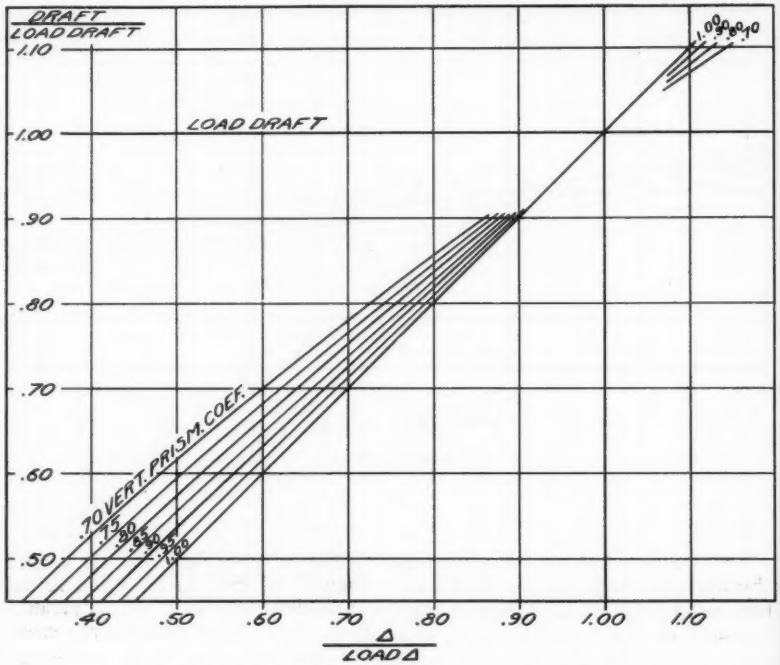


Fig. 17.—Variation of Displacement with Draft.

TABLE II

Fraction of Draft	Displacement Act.	Displacement Pred.	Tons/In. Act.	Tons/In. Pred.	KM Act.	KM Pred.	Displacement Act.	Displacement Pred.	Tons/In. Act.	Tons/In. Pred.	KM Act.	KM Pred.
1.00	17,500	17,500	59.8	59.9	29.0	28.8	5925	5925	29.1	29.1	19.8	19.9
.90	15,400	15,500	58.2	58.9	28.7	28.7	5275	5250	28.6	28.7	19.8	19.9
.80	13,450	13,500	56.8	57.8	28.8	28.7	4600	4600	28.2	28.2	20.0	20.0
.70	11,550	11,500	55.4	56.4	29.3	29.2	3975	3950	27.7	27.7	20.6	20.6

Type	Cargo Ship	Tanker
Prismatic	0.678	0.737
B/H	2.44	2.54
Vertical Prismatic	0.858	0.878
Midship Section	0.982	0.992
Actual GM_L	476	320
Predicted GM_L	483	336

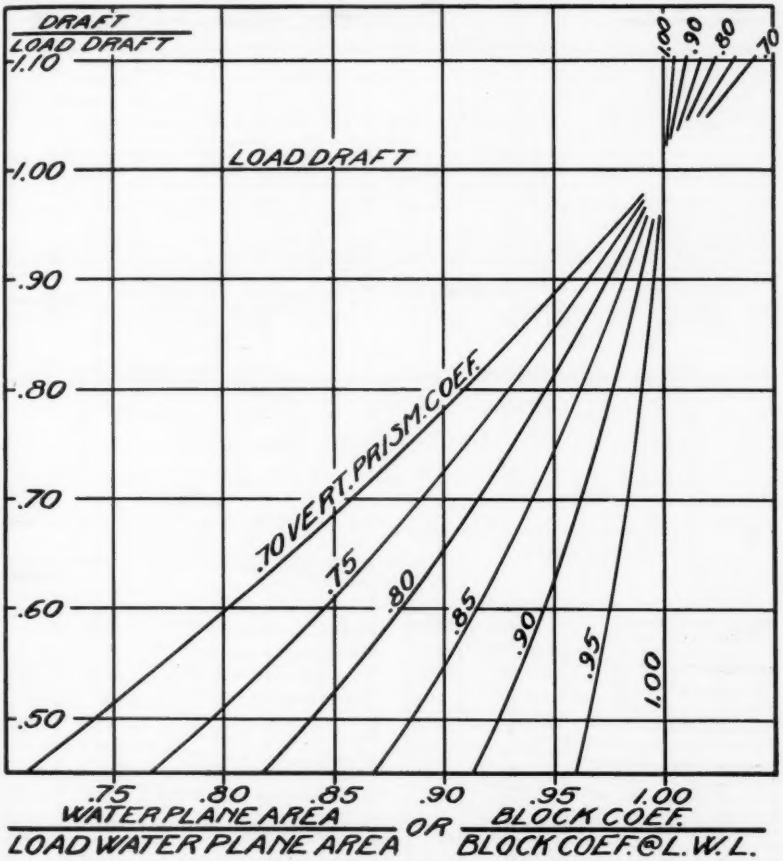


Fig. 18.—Variation of Water Plan Area or Block Coefficient with Draft.

curve; they are identical in form with Figure 12 and cover the range from 0.70 load draft to 1.10 load draft. This should be ample to include the point where minimum stability occurs in the usual liner or freighter. Note that the prismatics and beam to draft ratios are those for the load draft condition, and do not change with the particular draft under consideration.

No values are given below 0.70 of load draft because the accuracy of the transverse inertia coefficient, which were primarily given for the load draft condition in Figure 11, begins to fall off. However, a curve of KM plotted for the range given by Figures 12 and 19 through 22 should be ample for design purposes until lines are drawn and more exact computations made. Table II gives predicted values from the preceding figures and actual ones from the displacement curves for one large and one small ship. The actual vertical prismatic of each ship was used in determining the displacement and tons per inch variations.

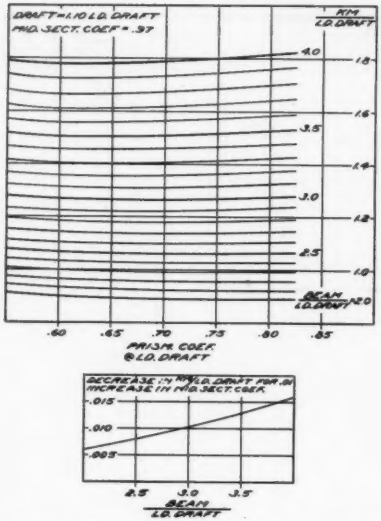


Fig. 19.—Transverse Metacenters at 110% of Load Draft.

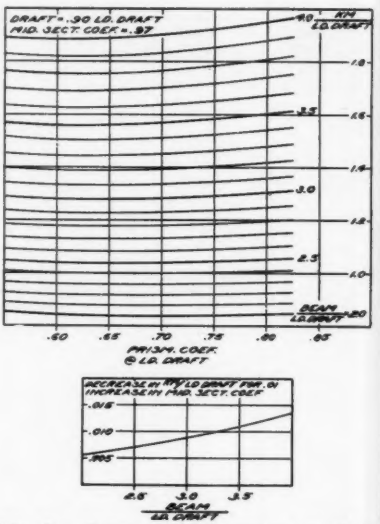


Fig. 20.—Transverse Metacenters at 90% of Load Draft.

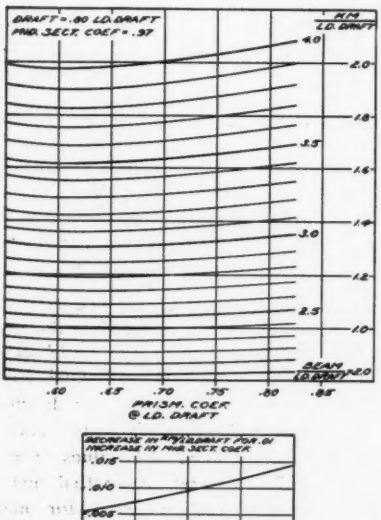


Fig. 21.—Transverse Metacenters at 80% of Load Draft.

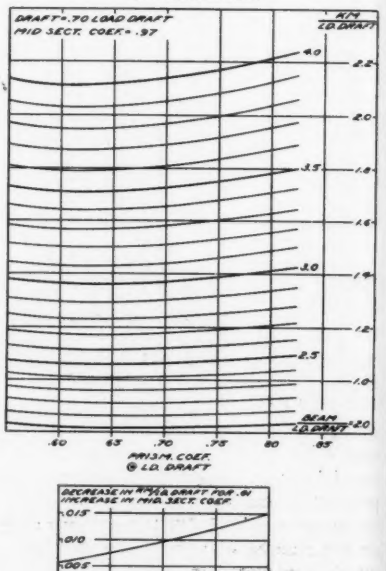


Fig. 22.—Transverse Metacenters at 70% of Load Draft.

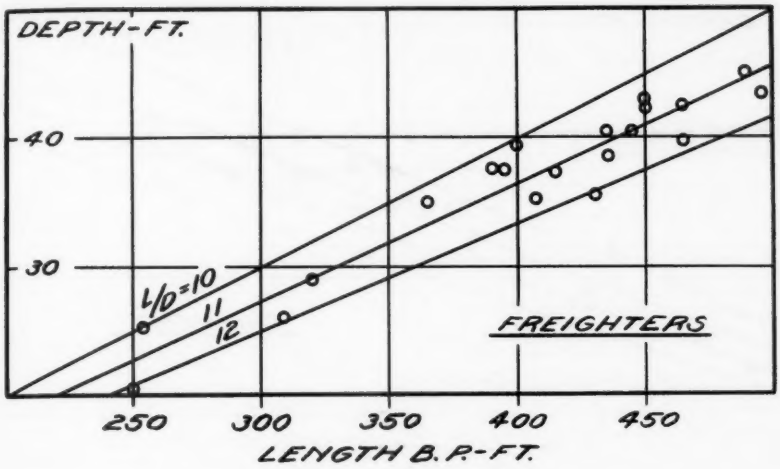


Fig. 23.—Cargo Ship Depths.

DEPTH

Depth is a rather complex variable in the design process, since it affects freeboard, cubic capacity of the hull, the location of the vertical center of gravity, longitudinal strength, and tonnage. Only the first two aspects will be considered here.

For a first approximation, Figure 23 shows that the normal American cargo ship has a depth to the uppermost continuous deck of 1/12 to 1/10 of the length between perpendiculars. Tankers, above 300 ft. in length, are somewhat shallower; Figure 24 indicates the fraction of length falling between 1/14 and 1/12.

Since draft is usually fixed in advance of depth, it is necessary to add at least the statutory freeboard, for a full scantling ship, with the upper tween deck height, in addition, for a shelter deck type of freighter. Figures 25, 26 and 27 may be used as rough guides for minimum depth. These have been prepared, in accordance with the Freeboard Rules[9] for freighters with standard sheer and camber and a block coefficient, at 85% of the freeboard depth, of 0.68 or less. For the shelter deckers, the tween deck height is shown and a tonnage well 6 ft. long is assumed. Note that the L/D values given are to the *upper-most continuous deck* and *not to the freeboard deck*, except for the case where these decks coincide. Thus, given length and draft, it is possible to read off the L/D ratio for the type of vessel under consideration and, tentatively, establish a minimum depth. For fuller ships, or those where non-standard characteristics are contemplated, the Freeboard Table corrections can be made to the depths shown in Figures 25 to 27.

The tanker case is considered, in an identical manner, in Figure 28. Here a value of 35% of length is assumed for intact superstructures. The chief variations from standard, in this type of vessel, are fullness and sheer; the former is almost always in excess of 0.68 and the latter less than standard,

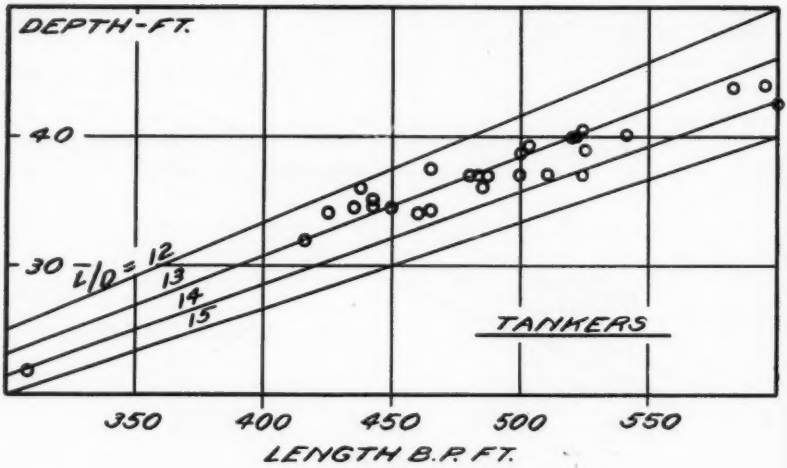


Fig. 24.—Tanker Depths.

and may be introduced only at the ends of the vessel, or omitted entirely. Approximate correction curves are given for the omission of sheer and increase in fullness. These corrections should be added to the designed draft before entering the first part of the figure.

Since the three-island type of freighter is almost past history, on this side of the Atlantic, it was not considered. Furthermore, because of the many variables in, and ramifications of the Freeboard Rules, Figures 25 through 28 should be used only for preliminary figures.

Depth also has an important bearing

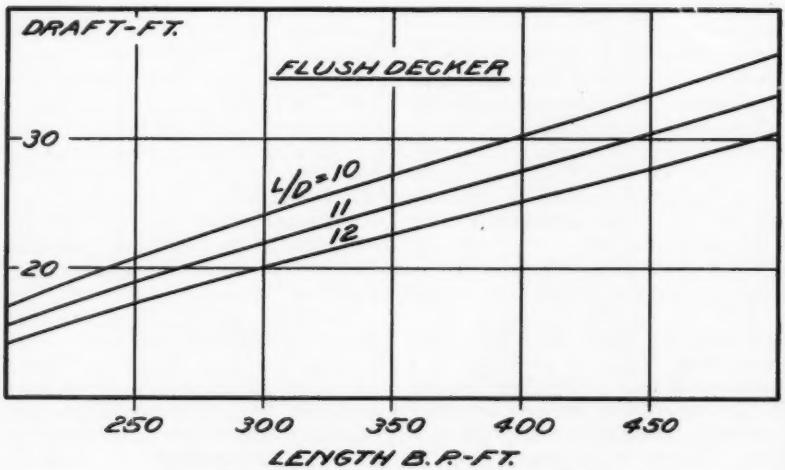


Fig. 25.—Statutory Drafts—Flush Deck Freighters.

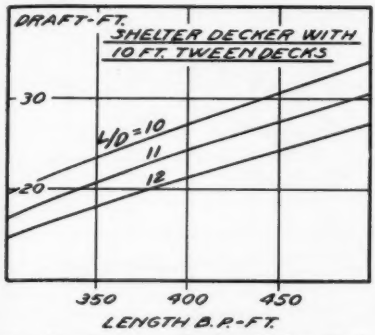


Fig. 26.—Statutory Drafts—Shelter Deckers with 10 ft. Tween Deck Height.

on cubic capacity and will, therefore, in some designs, be greater than would be required simply from a freeboard standpoint. It must be great enough to provide the necessary bale capacity, in a dry cargo ship, or tankage, in a liquid cargo vessel.

To estimate the former, the familiar bale coeffcient can be used:

$$C_h = \frac{V_h}{LBD_h}$$

C_h = bale coefficient.

D_h = molded depth of hold, upper deck to tank top, ft.

V_h = bale capacity of cargo holds, cu. ft.

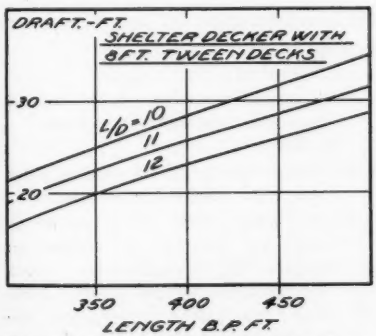


Fig. 27.—Statutory Drafts—Shelter Deckers with 8 ft. Tween Deck Height.

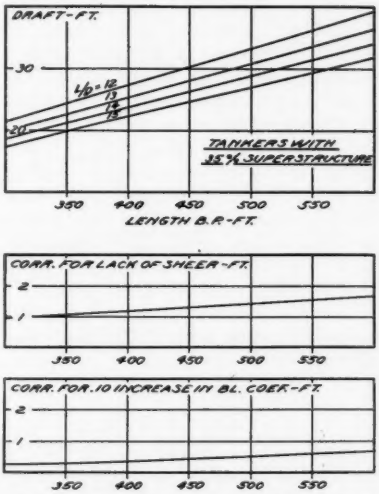


Fig. 28.—Statutory Drafts—Tankers with 35% Superstructure.

Figure 29 shows such coefficients for 15 cargo ships, of modern type, with a continuous weather deck over most of the length. The bale coefficient is normally 0.12 to 0.18 less than the block coefficient. Three island ships, with cargo in the superstructure, will have greater values than those shown,

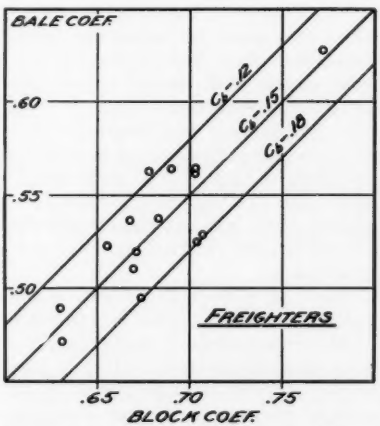


Fig. 29.—Bale Coefficients for Freighters.

when the depth is measured to the uppermost continuous deck; none of these are shown on the plot. Ships with a minimum of deep tanks, tween decks, bulkheads and other structural interference to cargo stowage, may, also, have higher values than shown. Three ships, of the fifteen plotted, carry cargo in a raised forecastle over No. 1 Hold, but, since these have either reduced, or no sheer, they show no particular variation from the vessels lacking such superstructure space.

The 25 tanker capacity coefficients fall in the same range as the bale coefficients of freights:

$$C_T = \frac{V_T}{L\,B\,D}$$

C_T = capacity coefficient.
V_T = gross volume of cargo tanks, cu. ft.
D = molded depth from baseline to upper deck, ft.

Lubricating oil tankers and others with numerous cofferdams, pump rooms, etc., will have lower coefficients than those indicated in Figure 30, as will vessels with extensive machinery spaces for direct-connected, slow speed diesels. Those with a long poop deck, as the upper boundary of the after tanks, together with short dry cargo holds and deep tanks forward, will show much higher values.

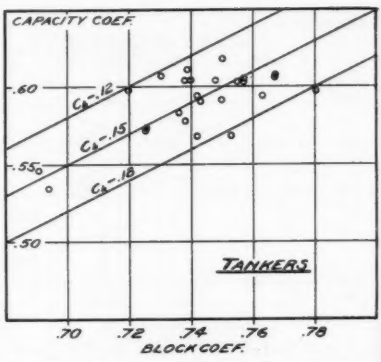


Fig. 30.—Capacity Coefficients for Tankers.

POWER AND REVOLUTIONS

In 1938, Yamagata presented a very ingenious method, based on numerous model tests made in Japan, for determining the shaft horsepower of merchant vessels with normal proportions and speed-length ratios from 0.55 to 0.95. The author has found it very useful, particularly because it includes the effect of R.P.M. on shaft horsepower. He has, therefore, redrawn the original plots, but has transferred the metric units into English ones for convenient use.

Figure 31 is similar to Figure 8 and represents average Japanese practice for the relationship of L.B.P., service speed and displacement. It was the genesis of, and was developed in an identical manner to, Figure 8 except that, instead of calculated effective horsepowers, shaft horsepowers from model tests were plotted, based on clean bottom, trial condition, and a designed service propeller R.P.M. of 120. Standard characteristics for this figure are as follows:—

1. A cruiser stern is used.
2. A single propeller of the airfoil type with streamlined sections. it fitted.
3. The rudder is of the double plate type with streamlined sections.
4. The longitudinal center of buoyancy is located as shown in Figure 32, which also gives length-displacement ratios. Note that the latter represents the length of a model, in feet, having a displacement of one ton.

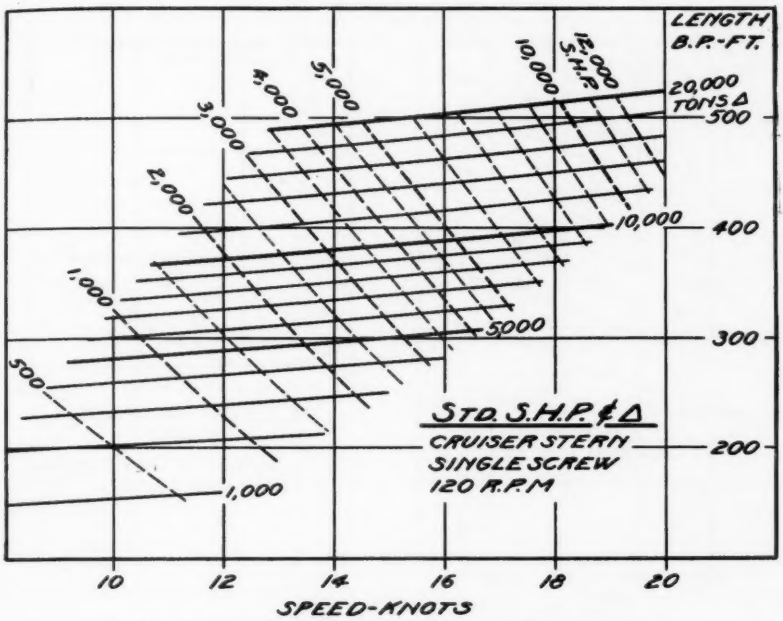


Fig. 31.—Standard Shaft Horsepower and Displacements.

The standard horsepower is read from Figure 30, for a given length and speed of ship, and modified by a series of factors to cover variations in the actual vessel under consideration.

$$S.H.P. = (S.H.P._o) K_a K_t K_s K_r K_d K_b K_r K_n K_p K_e$$

S.H.P. = shaft horsepower of actual ship.

$S.H.P._o$ = shaft horsepower of standard ship.

K_a = allowance for average bottom and weather conditions = 1.15-1.30

K_t = corr. factor for twin screw ships = 1.10.

K_s = corr. factor for a counter stern = 1.08

K_r = corr. factor for displacement = $\frac{\text{actual } \Delta^{\frac{2}{3}}}{\text{standard } \Delta^{\frac{2}{3}}}$

K_d = corr. factor for length-displacement ratio. Figure 33.

K_b = corr. factor for position of longitudinal center of buoyancy, Figure 34. Note that this is always greater than unity.

K_r = corr. factor for a contra-rudder = 0.97. This applies only to single screw ships.

K_n = corr. factor for service R.P.M., Figure 33.

K_p = corr. factor for a constant pitch propeller = 1.02. This applies only to single screw ships without a contra-rudder.

K_e = corr. factor for ogival section propeller blades = 1.02

Table III gives computed trial horsepowers by this method and model test results for a single and a twin screw ship. K_a is, therefore, omitted from the calculations.

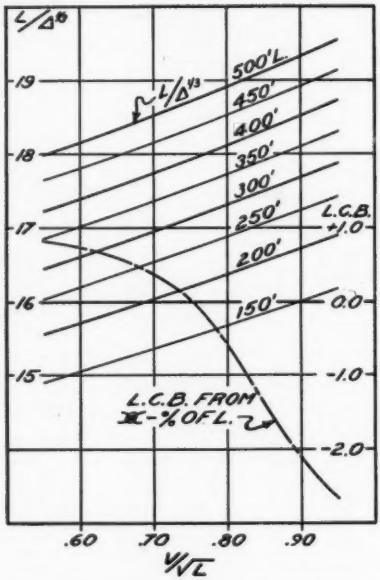


Fig. 32.—Standard Longitudinal Centers of Buoyancy and Length — Displacement Ratios.

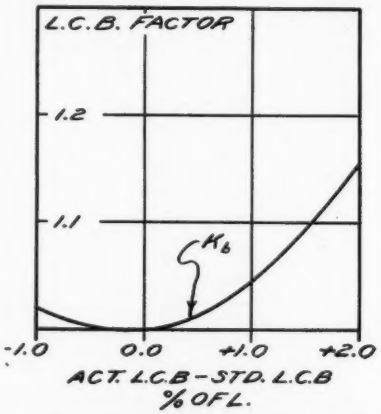


Fig. 34.—Longitudinal Center of Buoyancy Correction Factor.

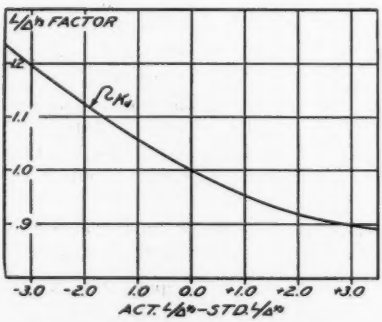


Fig. 33.—Fineness Correction Factor.

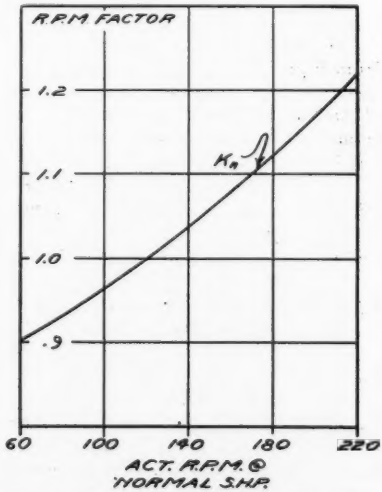


Fig. 35.—Revolution per Minute Correction Factor.

TABLE III

Item	SINGLE SCREW FREIGHTER			TWIN SCREW PASSENGER AND CARGO LINER		
	Actual Ship	Standard Ship	Correction Factor	Actual Ship	Standard Ship	Correction Factor
Length B.P., ft.	445	445		471.5	471.5	
Service Speed, Knots	15.5	15.5		16.5	16.5	
Displacement, Tons S.W.	14,024	14,600		14,027	16,600	
Number Screws	1	1	1.00	2	2	1.10
Type of Stern	Cruiser	Cruiser	1.00	Cruiser	Cruiser	1.00
$\Delta^{\frac{1}{3}}$	581	598	0.97	581	648	0.90
$L/\Delta^{\frac{1}{3}}$	18.45	18.25	0.99	19.6	18.5	0.95
L.C.B., % of L.B.P.	+1.07	+.17	1.04	−1.41	−.07	1.04
Rudder	Contra	Stream	0.97	—	—	—
R.P.M., Normal	92	120	0.95	90	120	0.95
Pitch	Varying	Varying	1.00	—	—	—
Blade Sections	Airfoil	Airfoil	1.00	Airfoil	Airfoil	1.00
Total Correction Factor			0.92			0.93
S.H.P.$_o$		5150			6750	
S.H.P. computed	4730			6270		
Trial S.H.P. from model tests	4650			6350		
% variation	+1.7			−1.3		

CONCLUSION

The author has endeavored to present, in graphical form, a number of approximations for use in preliminary design, particularly in the early stages, before drawings have been made and more exact computations can be undertaken. To do this, he has plotted data taken from existing ships, or completed designs, of recent vintage. Every effort has been made to make the resulting charts reflect, as nearly as possible, recent American practice.

It is well to keep in mind which ones of the foregoing figures are most basic and which are liable for revision from time to time.

Figures 1 and 2, covering deadweight-displacement ratios, show much higher values, today, than they would if drawn twenty-five years ago. Designers are continually striving to increase these values and will, no doubt, do so in future ships.

Figure 4 illustrates the current tendency toward increased beam for American vessels. Damaged stability requirements, higher centers of gravity resulting from welded construction, and extensive topside outfit weights, as well as occasional reductions in prismatic coefficients in conjunction with extra beam to avoid the sacrifice of displacement on a given length, have now made the beam about $L/9 + 15$ for our ships. Foreign practice, on the other hand, would show about $L/10 + 15$. The increased use of light alloys may tend, in the future, to prevent further beam increases.

Prismatic coefficients, Figure 5, while of basic importance, have shown little change since the advent of model testing.

Figure 6 reflects the use of more V-shaped and flaring sections in the lines

of modern ships and, hence, shows a higher value of load water plane coefficient, for a given prismatic, than a similar plot of a quarter century ago.

Figures 11 and 13, of transverse and longitudinal inertia coefficients, like Figure 5, will probably show little or no change with advancing years unless radical departures in hull form are introduced.

Figure 15, giving vertical prismatic coefficients, is derived from Figure 10 and will change accordingly. Figures 16 through 18, being simply plots of formulae, should remain static unless better formulae are invented or major changes in hull shape render present approximations invalid.

Depths, as given in Figures 23 and 24 together with capacity coefficients in Figures 29 and 30 are, already, highly variable quantities, from one ship to another. Both are interconnected with the complex freeboard and tonnage problems. Since the three-island type of freighter has been replaced, in this country, by a continuous-decked vessel, with or without a forecastle, depth has increased and over-all bale coefficients have diminished. Lighter, more compact, steam plants have improved tanker capacities.

In brief, then, the three charts showing the relationship of displacement, length and speed, Figures 7-9 are only valid if Figure 4 is valid. In the same way, the plots of transverse metacenters, Figures 12 and 19 through 22 are chiefly dependent upon the relationship of water-plane and prismatic coefficients given in Figure 10. Therefore, a continuous check should be kept on these two basic plots and the information deduced from them modified, from time to time, if serious deviations keep occurring.

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A NEW HIGH-PRECISION, UNIVERSAL-FREQUENCY COMPUTER COMPONENT

DONALD HERR, HONORARY MEMBER

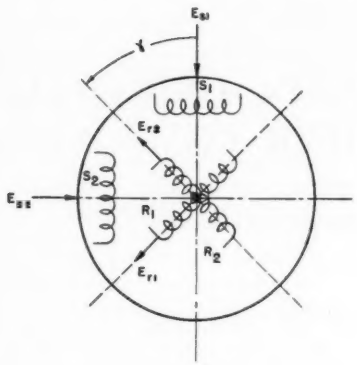
AND

NORMAN BLUMENSTOCK, NON-MEMBER

THE AUTHORS

MR. HERR, Honorary Member of the A.S.N.E., has the B.S. in E.E., M.S. in E.E., and E.E. degrees. He was National Coffin Foundation Fellow of the General Electric Company, National Tau Beta Pi Fellow, and National Sigma Tau Fellow at the Moore School of Electrical Engineering, University of Pennsylvania, and at M.I.T., prior to World War II. He was also awarded a National Gordon McKay Fellowship by Harvard University and received the A. Atwater Kent Award in Electrical Engineering from the University of Pennsylvania. A licensed radio amateur at 12, Mr. Herr first worked summers at RCA and Bell Laboratories and was with the General Electric Company in 1939 and 1940 as development engineer before volunteering for over five years of active Naval duty. He served as Officer-in-Charge, Electrical Minesweeping Group, Bureau of Ships, December 1940 to April 1943; as Acting Design Superintendent and Officer-in-Charge, Los Angeles-Long Beach Harbor Surge Investigation, U. S. Naval Shipyard, Terminal Island, to November 1944; and as Research-Patents Liaison Officer, Office of Naval Research, to January 1946, returning to inactive duty as lieutenant commander, U.S.N.R. Mr. Herr received two Navy letters of commendation. Since 1946, he was assistant to vice president, in charge of the engineering division of Control Instrument Company, Brooklyn, New York, and is project engineer at the Reeves Instrument Corporation, responsible for new servo and computer component developments. Mr. Herr has been associated with Dean Harold Pender and Professor Ernst Guillemin in advanced network theory and has specialized for 12 years in development and design of servomechanisms, differential analyzers, computers and fire control systems, utilizing advanced network analysis and synthesis methods. Mr. Herr is also presently teaching servomechanisms, network-synthesis and feedback amplifier design in the Graduate School of the Polytechnic Institute of Brooklyn. He has contributed frequently to the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS and was the Society's 1945 Prize Essayist on the subject: "Engineering in the Navy as seen by an Active Reserve Officer." He is senior member of the I.R.E., member of the A.I.E.E., A.S.M.E., American Mathematics and Physical Societies, New York Academy of Sciences and Sigma Xi, and appears in "American Men of Science."

MR. BLUMENSTOCK holds the degree of B.S. in E.E. from Pennsylvania State College. He has been with the Reeves Instrument Corporation for the past two years as an electronics engineer and is an associate member of the Institute of Radio Engineers.

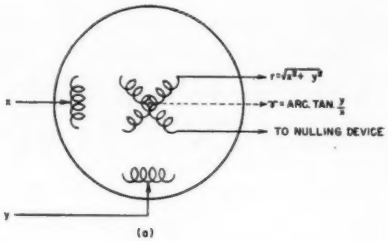


$E_{r1}(\gamma) = E_{s1}\cos\gamma + E_{s2}\sin\gamma$

$E_{r2}(\gamma) = E_{s1}\sin\gamma + E_{s2}\cos\gamma$

γ = ANGLE OF ROTATION

FIGURE 1 —RESOLVER INPUT AND OUTPUT VOLTAGES



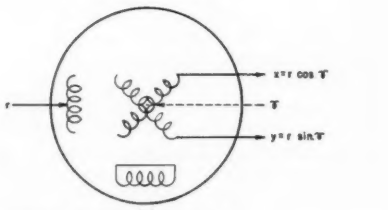


FIGURE2—RESOLVER FUNCTIONS

In any analogue-type fire-control system in which trigonometric operations are performed, a computing device known as the resolver is employed for such operations. Prior to World War II such a device was almost entirely mechanical, it being essentially a Scotch-yoke mechanism. These devices were large in size and weight, expensive to precision-machine, and required careful adjustment and maintenance in use. They were highly vulnerable to mechanical shock and vibration.

For approximately the past ten years, various embodiments of an a.c. electromagnetic resolver have been used to perform the essential trigonometric operations of rotation and transformation of coordinates, solution of plane and spherical triangles, and single-speed data transmission in modern fire-control systems.

Figure 1 depicts the basic operations which the a.c. electromagnetic resolver performs, and Figures 2a and 2b show two other derived uses of the resolver.

An a.c. electromagnetic resolver fundamentally consists of a stator and rotor, and, in one embodiment, the stator and rotor structures each contain a pair of distributed windings, each winding of the pair having its axes spaced 90° mechanical with respect to the axes of the other winding. This is also depicted symbolically in the above Figures.

Each stator winding is designed to generate a flux-density distribution in the air-gap between stator and rotor which is ideally sinusoidal in angle of position around the stator periphery, is a space-fundamental in the period of the air-gap circumference, and is proportional in maximum amplitude to the amplitude of the a.c. voltage applied to the stator winding.

Each such stator winding develops an induced voltage in one rotor winding which is sinusoidal, and an induced voltage in the other rotor winding which is cosinusoidal in the angle of rotor rotation with respect to stator.

Both rotor induced voltages have a space-period equal to the air-gap circumference and are proportional to the amplitude of the a.c. voltage applied to the stator winding.

Criteria of resolver quality are:

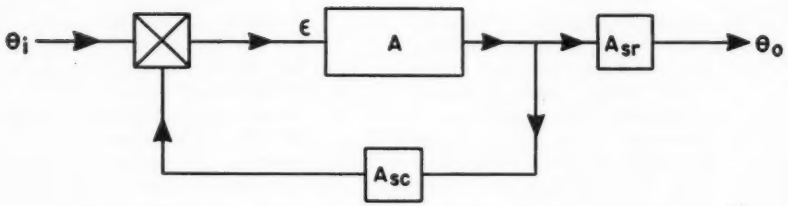
1. High angular accuracy;
2. Maximum degree of suppression of unwanted space-harmonics in the rotor output;
3. Maximum degree of suppression of unwanted time-harmonics and quadrature voltages in the rotor output, such unwanted components comprising noise;
4. Maximum input voltage range for which the resolver performance is of high quality (linearity with excitation, maximum signal-to-noise ratio, uniformity of input-to-output transformation-ratio);
5. Stability and reliability of operation;
6. Shockproofness and environment-proofness (temperature, humidity, corrosion);
7. Minimum size and weight;
8. Suitability for use in existing and future systems;
9. Universal-frequency operation with same physical resolver (to accommodate use at 60 cps, 400 cps, or 1000 cps by means of the same resolver);
10. Decreased unit cost compared with cost of all previous mechanical and electromagnetic resolvers.

Many factors enter into the design of a resolver suitable for use as a computing element. Some of the elements of design which basic research has elicited are:

1. Choice of a low core-loss, high-permeability, high-saturation-point, non-corrosive magnetic material;
2. Proper choice of numbers of stator and rotor poles and skewnesses;
3. Precision punching, proper handling, stacking, and insulation of stator and rotor magnetic laminations;
4. Controlled techniques of precision grinding of stator and rotor surfaces for maximum concentricity and annealing of magnetic surfaces;
5. Proper conductor distribution of stator and rotor windings, and insulation thereof, in the slots of the magnetic structures, for maximum suppression of unwanted space-harmonics (above the first, or fundamental);
6. Optimum dimensions of stator and rotor structures for minimized rotor inertia, maximized ratio of reactance-to-resistance of all windings, minimum slot and end leakage effects, and minimum variation in leakage inductance for all three common carrier-frequencies of 60, 400, and 1000 cps, in the same resolver;
7. Suitable techniques of controlled winding of stator and rotor conductor distributions;
8. Complete internal magnetic shielding of stator and rotor structures;
9. Precision bearing-mount of a continuously rotatable rotor;
10. Low-noise, low-resistance, slip-ring and brush rotor take-offs;
11. Mechanical rigidity of windings, resolver assembly, and case;
12. Accessible terminals;
13. Least weight and volume consistent with the foregoing.

Residual unwanted time-harmonic and time-quadrature voltages, variation in transformation-ratio, and the need for a high-impedance input (the resolver itself being essentially a low-impedance device) dictate the use of an accompanying compensating-booster-amplifier with the resolver for maximum performance quality as a continuous

GENERAL BOOSTER-RESOLVER-COMPENSATOR SYSTEM



A = BOOSTER AMPLIFIER TRANSFER-FUNCTION.
A_{sr} = RESOLVER STATOR-TO-ROTOR TRANSFER-FUNCTION.
A_{sc} = STATOR-COMPENSATOR TRANSFER-FUNCTION

$$\frac{\theta_o}{\theta_i} = \frac{A\,A_{sr}}{1+A\,A_{sc}} \longrightarrow \frac{A_{sr}}{A_{sc}}$$

FIGURE – 3

computing component. Its function and circuitry are shown symbolically in Figure 3 for the compensation of one stator.

Figures 4, 5, 6, and 7 depict a new resolver, now in quantity use, fulfilling all of the above criteria. Figure 8 shows a double-ended embodiment of the new resolver. Specifically, the component characteristics are given in Table I and its present performance characteristics, in conjunction with its accompanying booster-amplifier, are given in Table II.

It is noted that the Type R600 resolver is network compensated; the Type R601 resolver is winding compensated. Externally, Types R600 and R601 are identical except that the R600 has eight rear terminal pins and the R601 has twelve rear terminal pins (a total of four additional pins, two each for each of the added internal compensating windings). This is shown in Figure 9. It is to be noted that the new resolver is provided with a zeroing sector gear on its mounting flange for conveniently zeroing the resolver to the

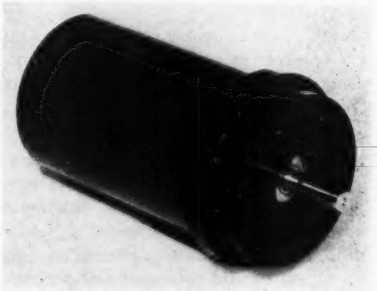


Fig. 4.—Physical Resolver.

TABLE I—RESOLVER COMPONENT CHARACTERISTICS

1. Universal Precision Resolvers—same resolver suitable for 60, 400 or 1,000 cps operation.
2. Maximum functional error of each stator-rotor axis is ± 0.05 of 1% over 360° of static rotation at all frequencies.
3. Minimum null and buckout voltages.
4. Minimum error in null-spacing.
5. Minimum inter-axis error.
6. Minimum variation of transformation ratio with excitation voltage.
7. Superior single-speed angle transmitter; maximum error 5 minutes of arc; r.m.s. error over 360° rotation less than 2.5 minutes of arc; zero minutes of arc error at 90° and 270° in sine function and at 0° and 180° in cosine function.
8. Superior precision phase-shifter when excited by two equal voltages 90° apart in time-phase.
9. The Reeves resolver is available for either network or winding compensation. This choice is made possible at all three frequencies by use of the appropriate compensating-booster amplifier. Type R600 Resolver is network compensated making it especially suitable for customary computer applications. Type R601 Resolver is winding compensated making it especially suitable for saw-tooth wave-shapes of high repetition rate.
10. Each compensating-booster amplifier will accommodate both stator windings of one resolver or one stator winding each of two resolvers.
11. Controlled machine winding.
12. Reeves Resolver is continuous-rotation type.
13. Single-ended or double-ended shafting.
14. The Reeves Resolver is fully electro-magnetically shielded internally.
15. All non-metallic parts are moulded, with high dielectric and mechanical strength.
16. Precision Barden bearings.
17. Rear pin-type connections in moulded assembly.
18. A zeroing gear is provided on the mounting flange for accurately and conveniently zeroing resolver to the system.
19. Volume of the Reeves Resolver is less than 18 cubic inches and its weight is approximately one pound. (one-sixth the weight and one-third the volume of existing resolvers having only one-half its accuracy).
20. Sturdy mechanical construction is maintained throughout.
21. Plug-in or permanently mounted booster amplifiers are miniaturized for minimum size and weight and maximum sturdiness.
22. Complete calibration and performance data are furnished with each resolver, obtained on highest accuracy specially-designed production tester.
23. Permanent construction for long life, yet resolver can be rapidly disassembled and assembled without deleterious change in performance.
24. Available singly or in quantity, with or without compensating-booster amplifier.
25. Low Moment of Inertia 0.75 oz. in^2; Low Frictional Torque 0.4 oz. in.

PERFORMANCE CHARACTERISTICS	REEVES R600 RESOLVER (Network Compensation)			REEVES R601 RESOLVER (Winding Compensation)		
Carrier Frequency C.P.S.	60	400	1000	60	400	1000
Input Impedance *	900 /72°	5400 /82°	15000 /78°	900 /72°	5400 /82°	15000 /78°
Transformation Ratio (Max.)*	0.960 /0°	0.970 /0°	0.970 /1°	0.935 /14° 0.950 /14° †	0.950 /2° 0.960 /2° †	0.950 /1° 0.960 /1° †
Input Impedance	2 MEG.	2 MEG.	2 MEG.	2 MEG.	2 MEG.	2 MEG.
Transformation Ratio (Max.)	1.0000 /0°0'	1.0000 /0°0'	1.0000 /0°0'	1.0000 /0°0'	1.0000 /0°0'	1.0000 /0°0'
Max. Functional Error over 360° Shaft Rotation (Direct Time-Axis Transformation-Ratio)	± 0.05 %	± 0.05 %	± 0.05 %	± 0.05 %	± 0.05 %	± 0.05 %
Rotor Null Voltage (mv/volt input)	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$
Null Voltage Principal Content	2 ND HARM.	2ND HARM.	2ND HARM.	3RD HARM.	3RD HARM.	3RD HARM.
Rotor Buck-Out Voltage (mv/volt input)	1	1	1	1	1	1
Buck-Out Voltage Principal Content	2ND HARM.	2ND HARM.	2ND HARM.	3RD HARM.	3RD HARM.	3RD HARM.
Steady-State Electrical Band Width (Min.)	± 5 CPS	± 50CPS	± 50 CPS	± 5 CPS	± 50 CPS	± 50 CPS
Error in Null-Spacing (Minutes of Angle)	± 1 $\frac{1}{2}$	± 1 $\frac{1}{2}$	± 1 $\frac{1}{2}$	± 1 $\frac{1}{2}$	± 1 $\frac{1}{2}$	± 1 $\frac{1}{2}$
Inter-Axis Error (Minutes of Angle)	± 3	± 3	± 3	± 3	± 3	± 3
Volts Input Range	0-20	0-20	0-20	0-20	0-20	0-20
Variation in Transformation Ratio With Input Volts (1/2 to 20 volts)	± 0.1 %	± 0.1 %	± 0.1 %	± 0.06%	± 0.06%	± 0.06%

* For Resolver alone. All other entries are for Resolver-Amplifier combination.
† Stator-Compensator Windings.
All entries are nominal unless denoted Max. or Min.

Table II.—Resolver Performance Characteristics.

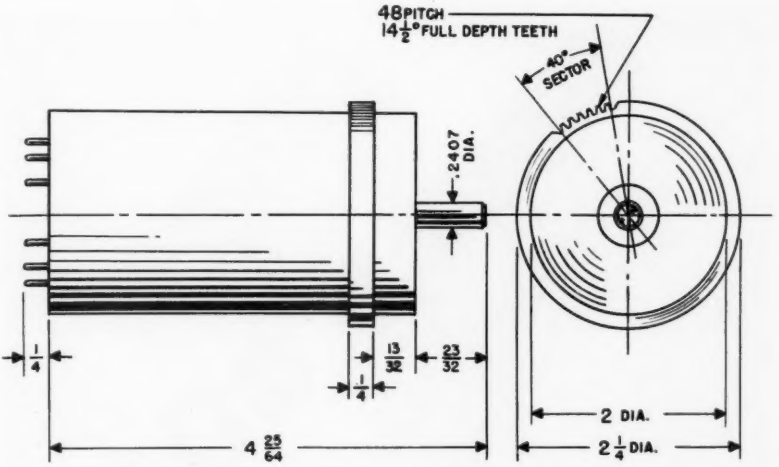


Fig. 5.—External Dimensions of Physical Resolver.

system in which it is used, rather than zeroing the system to the mounted resolver. Figure 10 illustrates the R600 resolver with its double-channel booster-amplifier. Figure 11 shows the R601 resolver with its double-channel booster-amplifier.

Figure 12 depicts precision stacking, cementing, and baking of stator and rotor laminated structures; Figure 13 shows automatic and controlled machine-winding of resolver rotors; Figure 14 illustrates production assembly of completed resolver units; and Figure 15 shows booster-amplifier development and production facility.

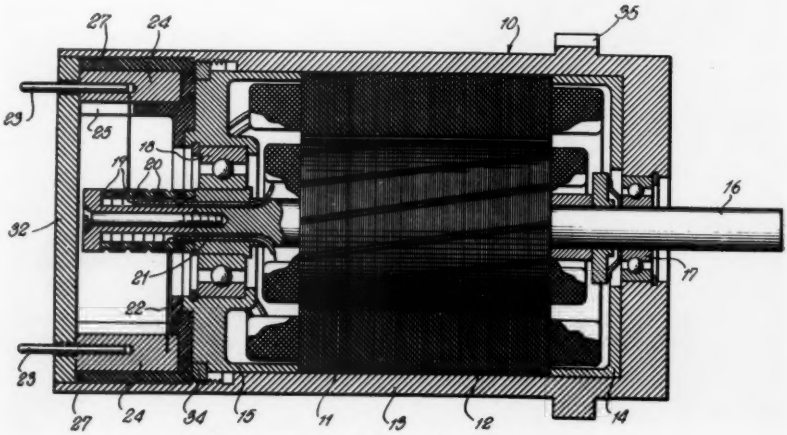


Fig. 6.—Cross-Section of Physical Resolver.

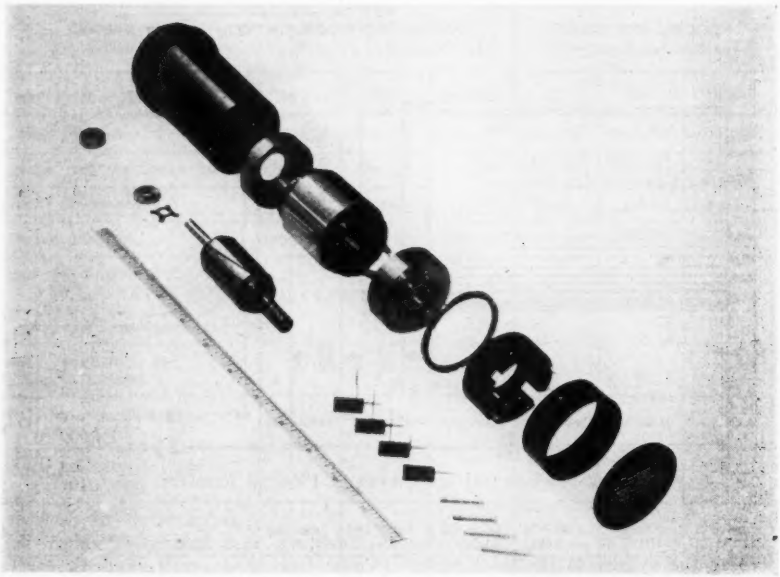


Fig. 7.—Exploded View of Physical Resolver Showing All Elements.

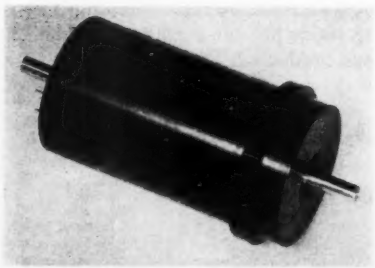


Fig. 8.—Double-Ended Resolver.

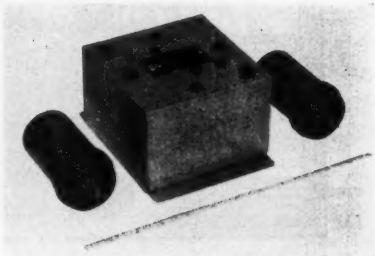


Fig. 10.—Type R600 Resolvers and Compensating Booster-Amplifier.

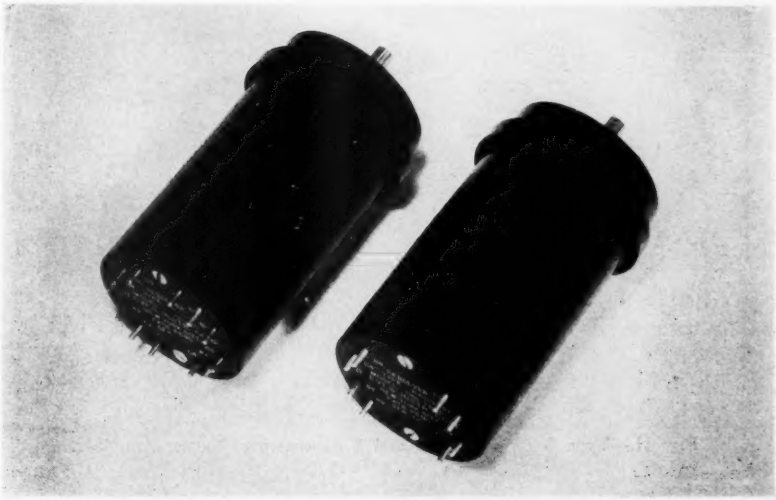


Fig. 9.—Type R601 Resolver (Left) and Type R600 Resolver (Right).

Fig. 12.—Precision Stacking, Cementing, and Baking of Stator and Rotor Laminated Structures.

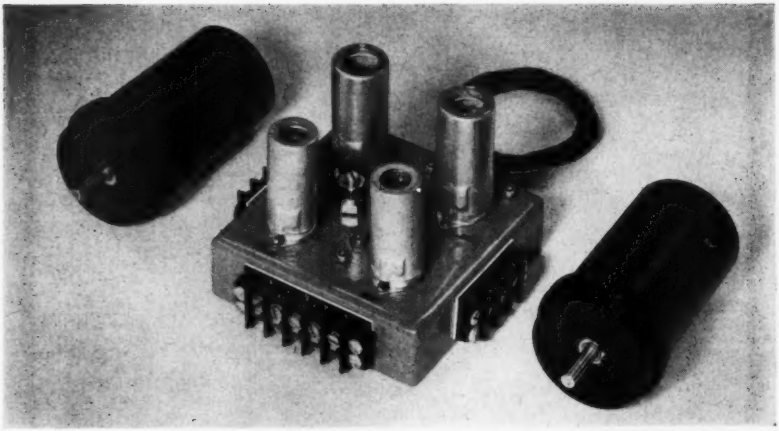


Fig. 11.—Type R601 Resolvers and Compensating Booster-Amplifier.

Fig. 13.—Automatic, Controlled Machine—Winding of Resolver Rotors.

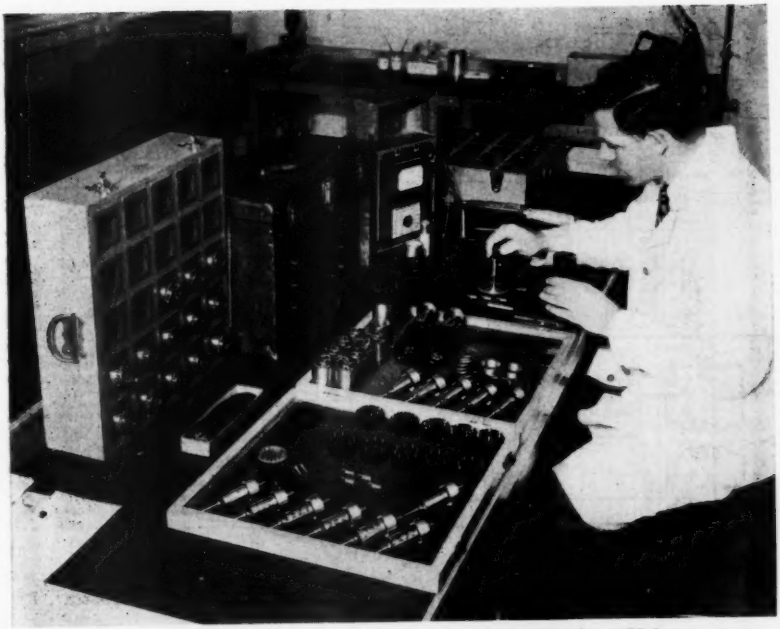


Fig. 14.—Production Assembly of Completed Resolver Units.

Fig. 15.—Booster-Amplifier Development and Production.

FIG-16 REEVES RESOLVER CHECKER (ELECTRICAL SYSTEM)

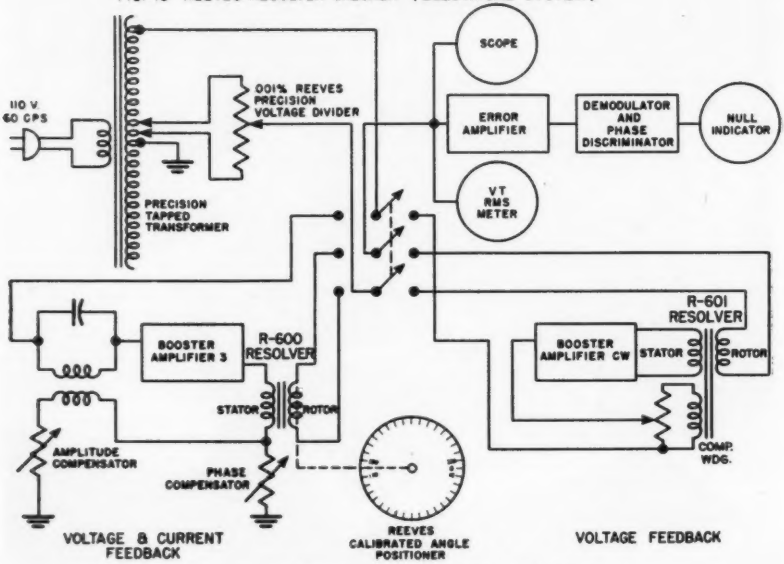


Fig. 16.—Reeves Resolver Checker (Electrical System).

In controlled manufacturing of a device of such high-precision performance, perhaps the most crucial, difficult, and expensive element is the provision of adequate and reliable test equipment maintained in a condition of less than 0.01 of 1 per cent error.

Figure 16 gives the schematic of the electromechanical resolver tester, and Figure 17 shows same photographically. Each resolver is mechanically positioned in angle by the precision calibrated gear-wheel and vernier appearing to the left, and the resolver direct time-axis rotor output is measured to five significant figures on the calibrated voltage-divider in the right central foreground of the tester. Such error measurements are made statically at every 10° over 360° of shaft rotation for each stator-rotor winding axis. In addition, the maximum transformation-ratios of the resolver, alone and with booster-amplifier, the d.c. and a.c. impedances of all resolver windings, and the null and buckout voltages (at 0°, 180° and at 90°, 270°, respectively) of the resolver and booster-amplifier combination are determined.

The basic new design of the R600 and R601 resolvers was confirmed as acceptable in each of the five prototypes made. Table III gives the complete technical summary of the first twenty production units made. Several hundred additional units are now in production, test, and system use.

This entire research, development, design, and manufacturing program, making available this new, superior resolver, was executed under Reeves Instrument Corporation auspices to fulfill urgent, long-term needs in meeting the increasingly stringent resolver component requirements of present and future computing systems.

REEVES PRECISION-RESOLVER PRODUCTION

TABLE III - SUMMARY

Resolver Serial No.	Stator Serial No.	Rotor Serial No.	Stator Mag. Matl.	Rotor Mag. Matl.	Type Resolver
2	120	121	C	C	R600
4	136	111	CA	A	R600
6	122	123	C	C	R600
8	132	137	C	C	R600
10	134	139	C	C	R600
12	106	105	A	A	R600
14	100	103	A	A	R600
16	102	101	A	A	R600
18	128	143	C	C	R600
20	130	125	C	C	R600
22	138	147	S	S	R600
1	104	107	A	A	R601
3	126	127	C	C	R601
5	108	117	C	A	R601
7	110	141	C	C	R601
9	112	131	C	C	R601
11	114	119	C	C	R601
13	116	115	C	A	R601
15	118	109	C	A	R601
17	124	113	C	A	R601
19	140	151	S	S	R601

C - Carpenter 49 iron-nickel alloy, hydrogen-annealed

A - Allegheny-Ludlum 4750 iron-nickel alloy, hydrogen-annealed

Compensation	Average Null (mv. at 10v)	Average Buckout (mv. at 10v)	Aver. Stator Zero Spacing-Min.	Stator Inter-Axis Spacing-Min.	Total Spread Percent Error
Network	5.0	18.5	1.5	1.5	±.05
Network	11.0	17.0	1.25	2.0	±.03
Network	6.5	15.0	0.75	1.5	±.04
Network	8.0	14.5	1.75	2.0	±.04
Network	8.0	17.5	1.5	1.5	±.05
Network	3.0	9.0	0.75	3.5	±.025
Network	8.0	17.0	1.25	2.0	±.03
Network	Magnetic	Material	Mechanically	Damaged	Unused
Network	7.0	16.0	1.25	1.0	±.035
Network	7.5	19.5	1.25	3.0	±.05
Network	6.5	25.0	2.5	0.5	±.035
Winding	9.5	10.5	1.75	3.5	±.04
Winding'	8.5	11.5	0.5	2.0	±.045
Winding	7.5	7.5	1.0	5.0	±.04
Winding	14.0	13.5	1.25	4.0	±.035
Winding	5.5	12.5	1.75	2.0	±.045
Winding	4.5	7.0	0.75	2.5	±.04
Winding	7.0	12.0	1.0	2.5	±.03
Winding	4.5	7.0	0.5	2.5	±.03
Winding'	4.0	9.0	0.5	1.5	±.03
Winding'	2.3	15.0	0.75	0	±.05

S - Electrical Silicon, "Transformer C," scale-removed

Winding - Stator and Compensating Windings Wound Together

Winding' - Stator and Compensating Windings Wound Separately

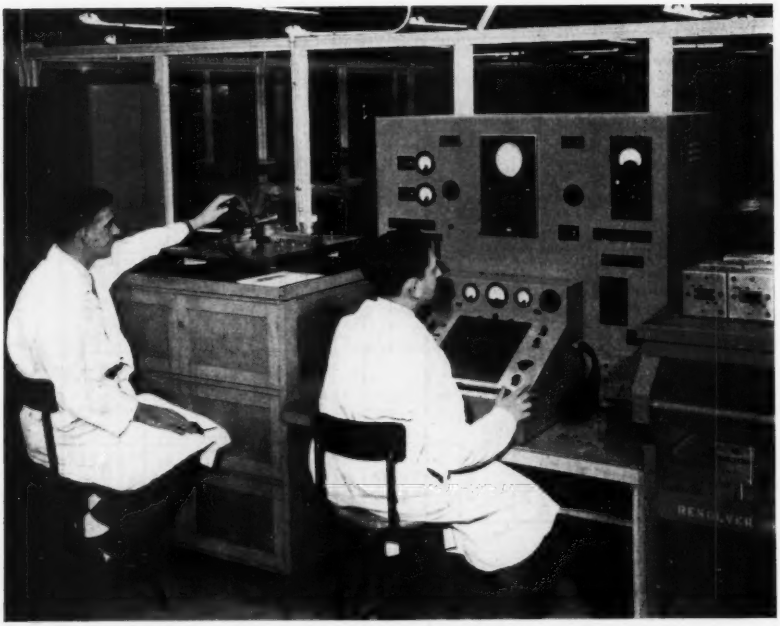


Fig. 17.—Precision Electromechanical Resolver Tester.

THE ART OF BILGE DIVING BROUGHT UP TO DATE

CAPTAIN ISAIAH OLCH, U.S. NAVY

THE AUTHOR

CAPTAIN ISAIAH OLCH graduated from the United States Naval Academy in 1922. A large part of his career has been spent in communications. He has written a number of professional articles but this is the first one to be published in the JOURNAL. The background for this article was obtained while Captain Olch was Senior Member of the Sub-Board of Inspection and Survey, San Diego.

The necessity "to see that the hulls of our vessels are kept in good condition" is as poignant today as it was when expressed in the opening paragraph of *"An Ounce of Prevention, or the Art of Bilge Diving,"* written by Captain C. W. Fisher (C.C.) U. S. Navy and published in the June 1933 number of the *U.S. Naval Institute Proceedings*. At that time the U. S. Navy had been relegated to third position as far as naval tonnage in world armaments was concerned; except for some submarine and intermediate cruiser construction, there had been no substantial replacement program for over twelve years; naval appropriations had been reduced drastically under the pressure of national economy and there was no intensive or long range progressive program underway in the research and development of new naval armaments.

Captain Fisher's article was most timely in that it appeared at a time when special emphasis was being stressed throughout the service on the introduction of damage control practices for our ships. Without pointing out specifically the reliance of successful damage control on structural integrity, the *"Art of Bilge Diving"* did bring to the attention of the operational and maintenance personnel the fundamentals for insuring such integrity. These were set forth as the progressive steps carried out in making a thorough material inspection to maintain the desired high standards of cleanliness, structural integrity and preservation.

Although marked advancements have been made in the art of naval construction since Captain Fisher's article was promulgated throughout the fleet in 1933, the basic requirement for structural integrity, watertightness, still exists and so does its worst enemy, corrosion. A considerable amount of time, effort and expense is still being spent in the endless battle against rust and although some of its troublesome sources have been removed by modern technology, a revision of Captain Fisher's article is in order to help fleet personnel in maintenance problems and to stress the important features of material inspections.

In contrast to conditions in the summer of 1933 ours is the leading Navy in the world today; our replacement of naval vessels is keeping pace with the requirements of our national policy; Congress shows a determined interest in providing for adequate naval appropriations and our operational development program looks far ahead into the future. To be sure that we do not rest on our laurels and fail to "see the ocean for the waves," we must maintain a steady advance in our ship upkeep standards along with the recent changes in naval construction and modern installations.

The most positive way to insure such high standards is to require the proper operation of all ship installations according to their designed performance and to carry out periodic material inspections to make sure that the ship's structural integrity is being maintained. To express it more bluntly, we must be sure that our Navy's material investment is being protected; we must not let deterioration of equipage or structure get ahead of us because replacements and repairs cost money which should be put to better use for training purposes. Routine positive operation of all machinery and regular material inspections followed up with prompt corrective measures can go a long way in keeping our fleet in material readiness. While the operational requirements for machinery installations are well covered in bureau manuals, the aspects of material inspections can be emphasized for the benefit of younger officers who have not had the experience in searching for material defects. In view of modern construction trends, a few pointers on where and what to look for on those Friday afternoon inspections may help keep our material standards up where the service requires them.

Catch the rust when it first appears; properly clean and prepare the surface; then paint it and the problem of corrosion is licked. That is as true today as when Captain Fisher expressed it. At that time manual scaling and painting were the Navy standard practices. Red lead was the number one standby as the old reliable preservative. Red lead gave way to zinc chromate and now more recent tests have indicated that a newly developed primer will replace the yellowish green of the zinc chromate. After what appeared to the poor seamen working over the side as eons of chipping hammers and after scornful looks from civilian industrial workers, power tools made their appearance to retire the scrapers and scaling hammers. With time the essence for rapid shipyard overhauls, sandblasting was put to use while ships were in drydock for work on the sides, bottoms and tanks. To still further ease the labors of structural upkeep, portable vacublasting rigs have been introduced for shipboard use in cleaning and preparing surfaces for painting. For the final surface protection, the paint pot and brush are slowly but surely giving way to the spray gun. Progress moves on and so does corrosion. It is still there ready to do its destructive work of weakening a ship's structure in spite of all the labor-saving gadgets. To catch rust when it first appears is still as important as ever.

Where and how to look for the telltale signs of corrosion were part of the theses of Captain Fisher's article. Modern naval construction has driven the formerly hidden signs out into the open and on the other hand, has added new places wherein it can carry on its unobtrusive destructive action. Where it was the practice to give minute attention to the appearance of rivet points, emphasis has shifted to a close examination of weld seams as the latter have come into more general usage as the art of welding progressed and the riveting hammer was discarded. Rivets have not disappeared completely—ship construction still makes ample use of them but instead of sounding the rivet points with a healthy blow of a testing hammer to dislodge any hidden rust, it

is now much easier to run a sharp screwdriver or marlinspike along a weld seam to reveal the corrosion.

In the prewar days of plenty of paint, structural rust sometimes remained hidden until the paint dropped off from its own weight! A protective paint covering approaching a quarter of an inch in thickness was not uncommon on even some of the smaller type ships. Many an engineering officer got grey in the service trying to figure out the fuel consumption necessary to transport all that extra weight around. Improved paint formulae plus the sad lessons of war have taught us to keep the paint thin. Scrape, clean and paint—once again repeat it to each new recruit, scrape, clean and paint; in that way costs will be kept down; fire hazards are reduced and maintenance is greatly improved because any subsurface corrosion can be readily detected and remedied. During inspections it is no longer necessary to chip off random sections of paint to look for rust—if modern painting standards are followed carefully, the rust will make itself known promptly.

It would surprise some inspecting officers if they checked with the bureau manual regarding authorized protective paint thickness—yes, it can be thinner than the proverbial gnat's whisker, so be sure that old paint is scraped away or well worn down to tolerable thinness before new paint is applied. If properly applied, it will protect against corrosion. However, if the latter does occur, then the thin paint will help the rust rear its ugly head so that corrective measures can be taken against it.

Linoleum no longer acts as a covering blanket for rust. Deck coverings are forever out as the accepted practice in surface combatant ships. The same goes for the nice wooden decks which used to cause the engineer officer still more grey hairs, worrying about the expenditure of fire brick for holystones. Maybe the teak was easier on the "pitch pounder's" feet than the present day bare decks are but at least the maintenance personnel know that no hidden corrosive danger of structural weakness is lurking out of sight. Next to wailing about the shortage of trained personnel following our hurry-up demobilization, a skipper's most plaintive lament was how to keep up the maintenance work on his decks. The answer to that one is as expressed before, "catch it while it's hot" and don't let the rust get started. But take heart because recent operational tests with vacublast equipment have proven the success of such sandblasting technique to warrant its adoption as standard equipment afloat for cleaning and preparing surfaces for painting. Follow up immediately with spray gun equipment and the labor problem can be reduced considerably.

Now then, as for the other hidden places, to some extent modern construction has added to them! Count up the number of bunk foot lockers on a man-of-war and it's no wonder that the First Lieutenant yearns for the good old fashioned jackstays and the swinging seabags. Inspecting parties should not fail to take a good look at locker bottoms as incipient dirt therein points to the potential danger of corrosion. The bottom spaces under the bunks in officers' country have no respect for rank either. The hidden areas under work benches and shelving in storerooms demand close scrutiny too. As for storerooms, a clear deck can mean a clear conscience for upkeep personnel because by keeping gear off the deck, even by the use of gratings, ventilation can be improved and sweating with subsequent danger of corrosion reduced. A clear deck area is also important for flooding in case of fire in a lower compartment.

Modern insulation may be neat in appearance when applied in large even bats but when inspecting for material upkeep, don't overlook the insulation securing buttons. The button stem goes all the way through to the supporting structure and a slightly yellowish tinge

on the button can be proof positive that corrosion is getting in its deadly licks in behind the insulation. Where this condition shows up most frequently in modern construction is on the overhead below the refrigerated spaces. Any marked discoloring of the insulation sheathing there means an item for the shipyard force at the ship's next overhaul.

Look under all foundations to uncover rust. That means under deck machinery, inside the false deck installed around machine gun or director mounts as well as the actual act of bilge diving in the engineering spaces to see how the undersides of the machinery pedestals or foundations look. Most of these places are hard to reach for proper preservation and that is where a bluejacket's ingenuity is called forth. If the deficiency is pointed out, some skinny shipmate with a long reach will be pressed into service and persistent follow-up on the inspecting officer's part will get the rust combated, no doubt about it. But it's his job to find the rust first and to find it, the inspector has to look for it. Only by carrying out searching inspections can our high material standards be maintained.

As Captain Fisher pointed out in his original article, the foregoing means that to properly hunt out the rust from all its hidden places, the inspecting officer is going to get his dungarees dirty. But didn't dungarees go out with coal burners? Unless he wants to ruin a good suit of working dress khaki, for carrying out a competent material inspection the officer should invest in some dungarees or cover-alls. Then he shouldn't be afraid to get on his belly and squirm into those hidden bilge pockets; if he can't get into them, he'll never know what their material condition is. Believe it or not but even those apparently inaccessible peak tanks in a DE or DD are built to permit visual examination in every pocket by a normally built officer. If inclined to be carrying a wee bit too much avoirdupois, a tour through even half of one of the peak tanks is highly recommended as a positive reducing measure—and for reducing rust too, so be sure to look at those overhead weld seams especially along the outboard boundaries and in the pockets around the rudder posts!

An accumulation of dust or dirt which is pushed inadvertently into angle iron boundary edges by deck swabs soon starts to act as an absorbing agent for moisture. In time that means the start of corrosion, so besides preaching that cleanliness is next to godliness just for the sake of appearance, start going after dirt to forestall the inroads of rust and subsequent structural weakness. Modern ventilation ducts are a far cry from the World War I type of fresh air conveyors; those 3" pipes sometimes let in the air, provided the cowl ventilators were trimmed right and there was a fair breeze to help matters along. The bluejacket today has little to complain about the volume of fresh air delivered into his berthing and working spaces; that is unless he has failed to keep the filters or heater fins free of dirt. Then the volume of air is blocked off and the dirt in turn can lead to the deterioration of the heaters and ducts. Access plates installed in the ventilation ducts should be used for the purpose of reaching in and cleaning out the system; the heater fins can be removed and cleaned to insure proper ventilation as well as structural maintenance. The large commercial type vacuum cleaner added to ships' allowances by recent general amendment is the answer for not only keeping the air ducts and heaters clean but also for reaching into those inaccessible bilge pockets. Housewives are not the only ones who enjoy the benefits of vacuum cleaners; shipkeepers too can take advantage of their labor-saving qualities but it is surprising to see how few ships have added the cleaners to their ship's allowance and put them to work.

Something new in our ships has been added for structural security along with heavy rust potentiality. The expansion joint is here to stay but its usefulness can be short lived if the inspection plates are not removed periodically to insure that the joint edges are clean and that no dirt has accumulated along the recess so as to allow an accumulation of water. At the first signs of rust streaks coming out of the bulkhead drain holes, open up the expansion joint deck plates and clean the bearing edges; the effort will be well worthwhile not only from the standpoint of maintenance but for safety of operation.

With linoleum gone from our ships, one other protector for rust was the deck covering in the wet spaces. Cement and tile in washrooms, heads, showers and galleys disappeared from wartime construction in view of the abrasive and shattering qualities of such covering as well as their shortcomings in watertightness. And so instead of testing the bond between the cement or tile and the deck by listening to the healthy bounce of a testing hammer, the inspector made a visual examination of the bare steel deck; needless to say he found plenty of rust along with a disproportionate amount of manpower being expended to check the rust. Thanks to modern advancements, an effective watertight binding covering is now being installed in our ships so that all wet spaces are not only more shipshape in appearance but they are serving to reduce considerably the time devoted to upkeep. But a tip to the initiated: when the new neoprene terrazzo is applied by the shipyard force, be sure that all equipment in the compartment is removed so that the deck is surfaced complete to the boundary edges. There is no improvement in maintenance or rust prevention if the working areas are surfaced and then water is allowed to drain off under ovens or working structures where cleaning is difficult.

Along with the improvement in deck coverings, crew morale has been greatly improved in modern construction by the introduction of more sanitary and readily maintained toilet facilities. The old constant flushing troughs and bucket basins handed down from apparently prehistoric times have finally been discarded. Instead of the former moisture laden and reeking atmosphere which contributed to excessive corrosion and countless man hours to keep up with maintenance, the copper-nickel and CRS accommodations of today with individual controlled flushing systems present more sanitary and serviceable conditions in keeping with the higher standards expected of our modern men-of-war. But another tip about that old demon, rust; where the wet spaces adjoin sheathed bulkheads, be sure that the insulation covering is well covered with a completely protective painted surface. If any cracks let the water seep through, then watch for a rusty discoloration along the bottom boundary of the insulation. If those brownish stains do appear, make out a repair card for the CSMP to have the area cleaned and re-insulated at next shipyard overhaul because those brown spots mean rust.

As for voids and cofferdams, in lieu of the former practice of opening them periodically and then retouching the surfaces after draining out all the water which had accumulated in the bottoms, present optimum maintenance practice is to make liberal use of dessicants for absorbing any moisture. In the "moth ball fleet" that is what is known as static dehumidification. Experience soon indicates the amount of dessicant to be used in each void and also the time for renewal; however, to be effective, absolute tightness of the area is essential, so when securing manhole covers to voids, take up evenly all around on holding down bolts or dogs to insure perfect gasket fit.

In prewar days when we took it for granted that all our ships were inherently stable, an avid inspecting officer would follow the premise of "A place for everything and everything in its place." With more and more topside

weight added as a result of the introduction of more AA armaments, radar and COC installations, the inspector should work according to the principle that "a use for everything and everything for its use" will be his safest guide. In other words, if it's on board ship then use it and if it has no use, then take steps to get rid of it. Keep all equipage in operating condition but if any equipment has become obsolete or has outlived its usefulness, don't wait for the Board of Inspection and Survey to start action toward its removal; start the ball rolling up the chain of command to the bureau to have it removed and thus save hours of manpower for needless upkeep as well as to improve structural stability through the removal of excess gear. Keep a ship's allowance filled as required for effective operations but discourage the "cumshaw" artists who gloat in getting excess gear on board ship just for the sake of hoarding. The most recent flagrant violators in this practice were the ships which serviced the targets before the Bikini tests; what they didn't take off the target ships wasn't worth taking or they didn't have any more room in which to stow the loot! And ever since then, vigilant inspecting officers have been plaguing the ships to discard the ofttimes worthless and excessive gear for the sake of ship stability.

Excess equipage with the attendant necessity for upkeep along with the potential dangers of overloading also applies to apparently useless supercargo in the form of needless stores. During inspections a questioning survey should be made of storerooms. Are they overstocked? Does the material appear to have been on board even before the ship was commissioned? Spare parts, of course, have their required and allotted space according to standard building and stowage plans, however, storerooms which have become overcrowded with useless or obsolete supplies as a result of oversight or misdirected zeal jeopardize the serviceability of the storerooms. They also present an added upkeep problem and in the case of combustibles, may become hazardous to the safety of a ship. Be sure that the storerooms are used according to designed purposes as shown in the ship's plans and then have the Supply Officer make periodic check-ups to insure the elimination of excess supplies; the inquisitive inspector can be greatly surprised at the boat loads of deadweight which can be removed and turned in to supply activities ashore by even the smallest type ship.

A couple of more tips while inspecting storerooms; keep material off the decks even if gratings are used to allow a scant half inch for ventilation and fire flooding protection, and insure that separated stowage is provided in the forward and after parts of the ship for the necessities of sanitation; the accidental flooding of a storeroom in which all the soap or head supplies were stowed could be of irreparable consequences until a return to port.

For the safety of your health see your dentist twice a year and for the safety of your ship take a good look at everything during inspection to insure reliability and freedom from hazards. Shipyards and industrial plants ashore emphasize safety to prevent injury and loss of man hours. Personal safety should be even of greater importance on board ship where trained personnel can not be replaced readily. Some of the dangerous items to watch for are smoothly worn ladder treads, unsecured work benches or gear lockers which could go adrift in heavy weather, improperly secured overhead weights which could be dislodged when underway, loose handrails on ladders and improperly illuminated areas during nightime operations. Miner safety type "hard hat" helmets are worn by practically all shipyard workers and yet personnel afloat do without them. Many a scarred skull could be spared especially down in the engineering spaces by wearing the small visor type. Next time your ship goes to the yard for overhaul, borrow a "hard hat" for the next round

of inspections; it will save many a bump when crawling around under the floor plates!

If the inspecting officer has reached that span where "life begins at 40," let him take heed when climbing through those long magazine ammunition trunks in tenders and the steep grades which lead up from tanker pump rooms. The use of a 21-thread manila line should be used as a safety measure; use a bowline around under the arms and have a man throw a turn of the line around a stanchion or a deck cleat while tending it as a preventer. A loose ladder rung or an inadvertent slip at the top of a forty-foot drop can be avoided by a few seconds taken in putting on a safety line.

The normal use of a flashlight during inspections can be improved upon as a safety feature. To avoid dropping it into an inaccessible bilge pocket, a half filled tank or even a gear casing where its removal might entail dismantling a part of the ship's plant, use a lanyard attached to the light. Splice a large loop at one end of the lanyard for securing around the neck and attach the other end to a ring secured to the base of the flashlight.

For safety sake check on the operation of all gear. Ask the man standing by how effective the operations are; the CSMP can grow progressively for the next shipyard overhaul as the thoroughness of your inspections progresses too. See that all moving parts are free of paint and that goes particularly for valve stems to indicate readiness for operation.

As the safety of a ship depends upon effective damage control, do not overlook all DC markings. Hatches, ports and cut-out valves should be plainly marked and legible markings should be on all piping to show at a glance the function and direction of operation of each line. A smart ship keeps all such markings up to standard and a vigilant inspector can keep them smart.

Elimination of wooden decking introduced more extensive use of glass fiber insulation in modern construction. The latter in turn introduced more hiding places for those pestiferous visitors from the beach, namely, vermin and roaches. In all food handling and sanitary spaces be sure that all sheathing boundary edges are tight as a safeguard against infestation. Knocking on the outside of the insulation bats will not dislodge roaches as was done before the war when bulkhead metal sheathing was mostly for appearance sake—the insulating material had worked itself down to the deck boundary limits a long time ago! Any small holes or cracks near the deck edge of the sheathing is an open invitation for the roaches to move in, so keep such damage well patched with tape and then painted both for appearance and effective sanitation.

As a Commanding Officer can not supervise personally the operation of all the equipage and machinery on board, he must accept as proof of its reliable and effective operation the normal fulfillment of the ship's routine. Only in case of accident or breakdown is his attention invited to any material deficiencies, in which case repairs are effected. If the serviceability of the installation should be changed, however, then an alteration is in order. But before dashing headlong to change any shipboard installation, remember that there's a purpose for everything on board ship and everything was put there for its own designed and allocated purpose. Before the machinery or the individual appurtenance was installed, careful thought back in the bureau concerned was given to the service requirements to warrant the installation in the first place. If its operational necessity afloat becomes obsolete due to changes in service requirements, the forces afloat should take the steps necessary to warrant the alteration. Make the recommendations if you think an alteration is necessary and the Type Commanders will soon take the ball from there on in to the proper bureau. But just because you don't like

the machine's characteristics, is no reason to junk it. If it's on board for a designed purpose, make it serve that purpose. Otherwise invite attention to its shortcomings or obsolescence with the view toward its removal from the allowance list. Remember the wartime adage that went: "if it moves, salute it; if it doesn't move, pick it up; if you can't pick it up, paint it"? Almost the same applies to the proper upkeep and maintenance of equipment on board ship in that "if you have it, keep it up; if you keep it up, use it as it's supposed to be used; but if it's not worth using, then take steps to get rid of it." Only do it officially; don't make unauthorized changes and that applies with great emphasis to structural changes.

Always remember that the various structural features of a ship were incorporated during construction to serve a specific purpose and any changes therein for personal convenience can endanger the integrity of the ship's structure as well as destroy the serviceability of the structure. Remember the pre-war days of the *Melville* and *Altair* and how through their long years of faithful service, their superstructures, like Topsy, grew and grew? If the exigencies of wartime service had not forced a drastic removal of most of their topside hamper, those ships would have approached the danger point where they would have rolled over in a heavy dew. Structural changes and the installation of superfluous equipage for personal desires should be avoided; during inspections check any changes in the ship's plans to be sure that the alterations have been approved by the cognizant bureau. The fact that the Admiral or Squadron Commander wants a new galley installed for the flag or desires a bulkhead knocked out to give the cabin more air is not authority for the change and he should not insist upon it without bureau authorization. What a kettle of fish would our ships be in if every new Admiral or skipper changed the ship's arrangement to satisfy personal whims! Operational necessity required the ship to be built that way in the first place but if the service requirements have changed to such an extent since the ship's construction to warrant a change, then the procedure as set forth for requesting authority for the change should be followed. The uninitiated may think that is the "red tape" way but service experience has shown it to be the most effective and economical way; rather that way than to have everyone go hog wild in making changes. However, inspecting officers should not frown on suggested changes as offered by operating personnel; observe and listen to the arguments both for and against the alterations, then if they are considered to be reasonable in view of service operating requirements, forward the recommendations on for proper review and investigation. In that way we can obtain a higher operational standard.

In carrying out inspections of other ships in connection with the competitive year or training cycle, do not be afraid to invite attention to any material deficiencies or improper maintenance practices. Failure to set our standards high will lead only to structural deterioration and operational failures, so point out the defects without fear that you will be black-balled out of the "mutual admiration club" when the other officers come over to inspect your own ship. The general efficiency "meat ball" can not be earned without a searching inventory of all material features, so set the standards high and use inspections to keep them that way.

The proof of proper maintenance, of course, is in the effective operation of the ship. Machinery and material histories on the other hand, can give a comprehensive picture as to the progressive and routine phases of such maintenance. The records can also substantiate reasons for necessary repairs or alterations, so keep all records and logs complete. In the event of breakdown or when approaching a routine shipyard overhaul, they can be of inestimable

value in having necessary repairs or changes made to keep the plant and the structure of the ship in the best of material condition.

If machinery did not wear out or break down; if ships' structures did not corrode and if all hands saw to it that dirt did not accumulate on board ship, maybe there would not be any need for routine material inspections. Likewise if all inspecting officers were conversant with the standards as set forth in bureau material manuals and if Commanding Officers permitted no unauthorized alterations to be made afloat, then there would be no need for the Boards of Inspection and Survey to make periodic inspections; the Boards could then devote their time to surveys of unserviceable ships and to supervise acceptance trials for new naval units. With only serviceable craft retained in the post-war fleet and but little new construction in the offing, the Boards could then be disbanded; except for the fact that their presence is required by law.

While routine machinery tests and periodic material inspections are directed by bureau manual instructions to insure the maintenance of operational standards for reliable service requirements, it wasn't until 1882 that the law of Congress established the requirements for the Board of Inspection and Survey. Prior to that time, as new naval construction was undertaken, officers were designated to supervise the building of our ships and then to observe the builder's trials to warrant acceptance of the ships which met the desired service standards.

President Washington could be described as the first member of a Board of Inspection and Survey as he paid several visits to the *United States* during its building to observe personally the progress of construction. Before the advent of steam, however, preliminary inspections were made to ascertain the quality of material used during construction as conforming to specifications.

In 1860 a Board was appointed to review the practicability of converting all our sailing vessels to steam. However, the war came along and although the Board was prevented from completing its study, steam was introduced into the Navy—a favorable exigency of the service under the stress of war. The first acceptance trial requirements as laid down by the Navy might be said to have been those covering the service tests of the *Monitor;* they covered the optimum test which might be required of any man-of-war, namely, operations under fire by the enemy. As expressed in the terms under which the *Monitor* was built: "The Government requires ninety days in which to test the vessel. So soon as the vessel is ready for service, the Government will send her on the coast and put her before the enemy's batteries in the service for which you intend her. No other test can be made to prove the vessel . . . The plan is novel, and because it is so, the Government requires the designer to warrant its success. Placing the vessel before an enemy's battery will test its capacity to resist shot and shell . . . The Government cannot consent to receive the vessel until she shall have been tested in the manner proposed." The agreement with the builder did not cover the presence of a Board of Inspection and Survey on board during the acceptance trial and the structural tests were carried out long before the proving ground was set up at Dahlgren!

With only three navy yards along the east coast in competition for the allotment of funds to carry on ship repair work, Congress in 1882 passed a law which read as follows:—"(August 5, 1882; 22 Stat. 296-297, Chap. 391) It shall also be the duty of the Secretary of the Navy, as soon as may be after the passage of this act, to cause to be examined by competent boards of officers of the Navy, to be designated by him for that duty, all vessels belonging to the Navy, not in actual service at sea, and vessels at sea as soon as practicable after they shall re-

turn to the United States, and hereafter all vessels on their return from foreign station, and all vessels in the United States as often as once in three years, when practicable; and said boards shall ascertain and report to the Secretary of the Navy, in writing, which of said vessels are unfit for further service, or, if the same are unfinished in any navy yard, those which can not be finished without great and disproportionate expense, and shall in such report state fully the grounds and reason for their opinion. . . ."

With but few ships in service at that time, the members of Congress who represented the navy yard sections could demand a prompt accounting of the Navy's overhaul budget requirements. A fair distribution of work to the yards could not be avoided with such a law in effect. But as the Navy grew in size and more navy yards were established along both coasts, the original precepts which prompted the law were lost sight of. However, the practicability of the Boards of Inspection became more pronounced. As long as the statute remained on the books, ships continued to be inspected every three years and whenever they returned to the States from extended periods of foreign duty. As ships usually were scheduled for an overhaul after such foreign duty, the inspection served as a pre-overhaul one to uncover deficiencies. In time it has become the practice for ships to be scheduled for examination by the Board of Inspection and Survey from three to four months prior to routine overhaul. Ample time is then afforded for the Type Commanders and bureaus concerned to review the inspection report; approve and authorize any alterations as recommended by the Board; allocate funds to the Shipyard for the work and to enable the overhauling activity to plan and assemble material for the alteration. For a ship to obtain optimum results from the searching inspection carried out by the Board of Inspection and Survey, the inspection should be scheduled well in advance of the shipyard availability.

The legal importance of such an InSurv inspection is not generally known to personnel afloat; according to the law which established the Board, the bureaus concerned are required to explain what steps are proposed to correct any material deficiencies or to carry out any design changes as recommended by the Board of Inspection and Survey. In other words, if InSurv says it must be so, the bureau has to let the front office know why it can't be or what steps are being taken to make it so. If a training cycle material inspection is due and at the same time a yard availability is not too far away, by all means have the Board of I and S officers do the job and get the benefit of their recommendations.

Aside from the legal backing of their report, such an inspection is more beneficial from the over-all service standpoint because as the result of going through all the various types of ships, the Board of Inspection and Survey has had a better picture of naval general maintenance standards. The Board's officers, from their experience, are in a better position to point out deficiencies, violations of safety practices and to advise as to proper maintenance standards. Operating directly under control from the Chief of Naval Operations the Board is in a better position to give a fair and impartial appraisal of a ship's standards than would be done by a Board from a sister ship in the same force. So if the Board of I and S is scheduled to inspect your ship, welcome it on board with open arms and rest assured that it will call a spade a spade with great benefit to the ship and the service as a whole. Observe how the Board practices the "art of bilge diving" with the view toward making future inspections just as thorough and searching so that our material standards will remain as high as possible and thus assure the protection of our Navy's material investment.

A MERCURY SHAFT SEAL*

JOHN W. SAWYER AND LYNDON CRAWFORD

THE AUTHORS

LYNDON CRAWFORD attended Harvard University for three years and transferred to Massachusetts Institute of Technology. He received a B.S. degree from the latter institution in 1938 in the Department of Naval Architecture and Marine Engineering. From 1938 to 1945 he was employed by Newport News Shipbuilding and Dry Dock Co. largely in the Hull Technical Division of that company. In 1945 he moved to The Glenn L. Martin Company where he served as a senior stress analyst. In 1947 he became associated with the S. Morgan Smith Company in its Special Products Section. He joined Reed Research, Inc., in 1948.

JOHN W. SAWYER graduated from N. C. State College in 1933 with a B.S. degree in E.E. He then served a number of years in the merchant marine and later received a B.S.M.E. degree from Ohio State University. In 1942 he entered the Navy. Since release to inactive duty, he has been engaged in gear development work in the Turbine and Gear Branch, Bureau of Ships, Department of the Navy. He devised the basic seal described in this article.

1. Introduction

The seal problem is an old one. Wherever two mutually separate compartments have been pierced by movable members the demand for establishing an effective seal has existed. The seal served a primary function of prohibiting flow of fluid from a high to a low pressure area. A critical requirement was that the means of establishing a seal be such as not to result in undue friction losses and damage to moving elements. With high pressure differences, large moving parts and high relative speeds the matter becomes increasingly important.

The usual answer to the problem has been to provide a particularly long or difficult passage tending to prohibt fluid flow. A smooth shaft entering a pump housing through a conventional packing gland is sealed by the narrow passage between shaft and packing. A labyrinth seal provides a long and difficult passage which also inhibits fluid flow. This paper discusses a seal which opposes fluid flow by means of an internal pressure field.

In the first applications of seals pressures were quite low when compared with those found in general practice today. Relative speeds, between the rotating or reciprocating element, the

* Patent Pending.

shaft or piston and the packing or housing, were considerably less than those encountered in current practice.

Early problems relating to the sealing of a medium in a compartment, such as a pump cylinder, containing a moving element actuated by a shaft, were without doubt as difficult to solve, relatively, as those encountered in engineering applications of today. Materials, production equipment and methods of 30 or 50 years ago were not comparable to those of today. The balancing features, however, were the lower pressures and relative speeds that seals were required to meet. Thus it would appear safe to assume that the seals required in 1900 were developed with the use of existing materials as readily as present day problems of sealing are resolved.

With the advent of high shaft speeds and increased operating pressures the demands for satisfactory sealing under varying degrees of radial and axial misalignment still remained very real.

Seals for relatively large shafting, operating under conditions of high rubbing velocities, present a number of rigid requirements that are regarded as essential for satisfactory performance. These include such factors as high accuracy of manufacture, careful installation, lubrication of surfaces, reduction of heat losses, ease of maintenance, dependability and alignment under operating conditions.

During recent studies of a shaft seal problem the use of a centrifugal mercury seal, supplemented by a rubbing seal, which was to function only at low speeds, was conceived as a possible solution. This paper undertakes to describe certain outstanding problems pursuant to this device; in addition a test model and experimental work are described.† The state of present knowledge together with a study of possible specific applications to shafting, pumps and compressors is set forth in a general manner.

II. The Seal

This seal is referred to as a mercury seal because it utilizes a mercury charge to establish the seal under normal conditions. The principal of the seal is schematically illustrated in Fig. 1. Details are limited in this Figure to permit illustration of the basic seal construction and the principles of operation.

The seal consists of 4 main elements: namely the annular channels, (7)-(8), expandable elements, (2), static element (4) and the sealing medium (6) which is mercury. The shells bounding the annular channel are secured to the rotatable shaft through suitable means. Expandable elements are attached to and form a part of the channel. The static element is fixed to the housing. The sealing medium is mercury and is retained in the annular channel. Assembly and charging with mercury will be discussed later.

The steps in the operation of this unit are entirely automatic and do not require the use of any electrical or mechanical actuators. The various elements operate over the desired speed range and are actuated by the centrifugal forces developed by rotation. Under the condition of zero shaft speed the mercury charge remains at the bottom of the annular channel as shown in Fig. 2 position "o". Over the speed range o-c, Fig. 2, the flexible units grip the static element and thus prevent the passage of a medium such as water from one side of the seal to the other. With rotation of the shaft centrifugal force tends to lift the expandable element from contact with the static member. However, the design is such that centrifugal force on the mercury causes it to be uniformly distributed around the annular channel well in advance of the actual break in contact between flexible and static seal elements. This condition is reached at speed position "c", Fig. 2. Therefore, there is a speed range, b-c,

† Development work was conducted by Reed Research, Inc., Washington, D. C., under contract to the Bureau of Ships, Department of the Navy.

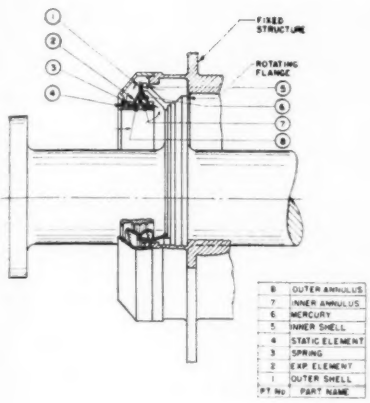


FIG. 1 MERCURY SHAFT SEAL - ILLUSTRATIVE MODEL

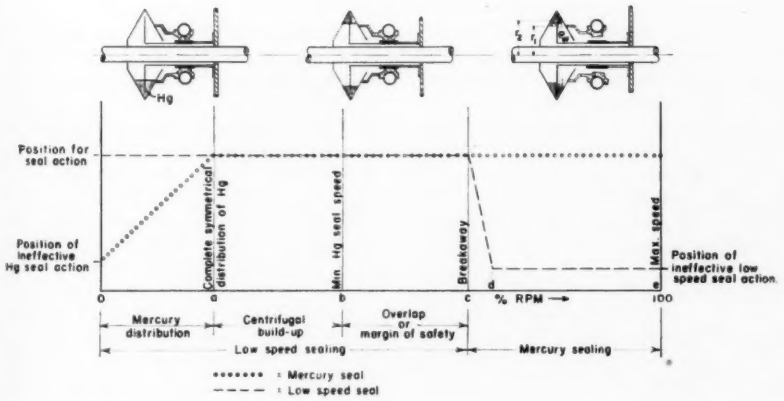
relatively narrow, in which there is a double or overlapping seal between the flexible and static elements and by the mercury. This condition, with the double seal existing, is illustrated in the range b-c. As the shaft speed is further increased beyond point "c" the centrifugal force lifts the flexible units from contact with the static member and the seal is then maintained by the mercury through the range c-e.

The curve in Fig. 2 indicates the positions at which the various elements operate with an increase in shaft speed from zero to maximum. In bringing the shaft from maximum speed to rest, the steps are reversed going from "e" down through d, c, b, a and o in order. Therefore, it is seen that a seal is maintained under all operating speeds from zero to maximum.

There are a number of features of this mercury seal that appear to be different from the seals available in normal commercial use. They include:

1. Low manufacturing tolerances required.
2. Elimination of contact between solid elements at high rubbing speeds.
3. Sealing under radial and axial misalignment.
4. Low friction losses.

FIG NO. 2
MERCURY SHAFT SEAL
SCHEMATIC OPERATION



III. Engineering Problems of Design

A. MERCURY SEAL COMPONENT

As previously explained, under normal conditions mercury provides the seal action. With steady operating conditions it can be expected that the mercury will tend to rotate as a rigid body, impelled by the frictional traction of the rotating shells. Some speed lag is to be expected due to frictional action within the body of the mercury, especially in the portion adjacent to the static disc. Thus the lag in mercury must be considered in designing for sealing of a given pressure.

Under the action of fluid pressure on one side of the mercury, the right hand surface of the mercury as shown in Fig. 2, it can be readily derived that the relationship between the radii of the two surfaces, the left and right hand Fig. 2, is as follows:

$r_1 = \sqrt{r_2^2 - 2p/\rho\omega^2}$ where:

r_1 — radius of non-pressurized surface in inches

r_2 — radius of pressurized surface in inches

p — pressure on surface at r_2, including centrifugal water effect-psi

ρ — mass density of mercury lb sec^2/in^4

ω — rotational speed of *mercury* radians/sec

The above equation can be utilized to yield a minimum speed at which mercury sealing is possible with a given design or a minimum pressure for a specific configuration. It is assumed that seal action is impossible when the large radius, r_2, exceeds the outer radius of the static disc. The actual shaft speed would be in excess of the mercury speed at this stage by a certain lag factor, which in early parts of the study was predicted as approximately 35 per cent of the shaft speed.

The problem of providing sufficient volume within the two rotating shells for mercury under the static condition should be appreciated with special reference to its inertia in starting.

Particular attention was given to the selection of nonamalgamating materials for the shell, static disc and any other parts likely to come in contact with mercury during any phase of the operation.

B. LOW SPEED SEAL COMPONENT

While this device can be most properly termed a ***mercury seal,*** actually the most critical element in the design is the low speed seal component.

Fig. 3 shows the elements in greater detail as applied to a model. Geometry is an important factor. Due to the lack of elastic predictibility of the collar (8), Fig. 3, it was felt that this element should serve only to provide flexibility. Since it drives a rather heavy spring under considerable preload, any design must be a compromise between the desire for optimum flexibility and, on the other hand, both torsional stiffness and dynamic stability.

Material considerations are also important. The contradictory requirement of low-friction rubbing between rubber and a fixed metallic surface was actually the first problem to be undertaken in the study of this device. That this is a first order matter, becomes apparent when it is considered that the initial contact pressure must be quite high; this low speed seal must remain effective at least up to the speed corresponding to minimum mercury seal. The low speed rubber element, to be effective, should resist wear under continuous operation and should attain only moderate temperatures if sustained operation at

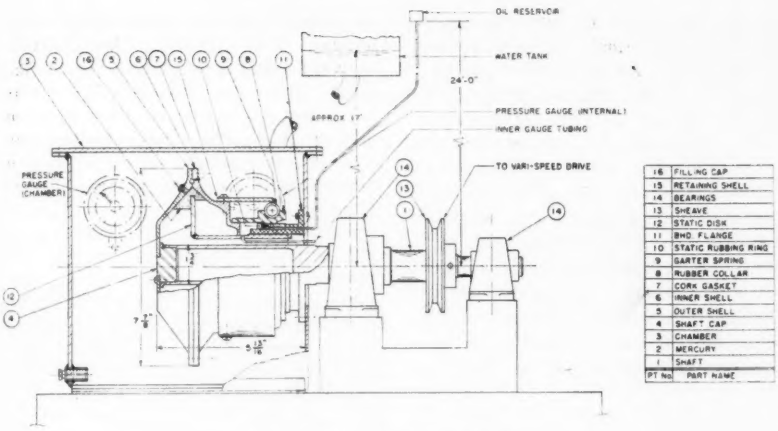


FIG. 7. SHAFT SEAL DEMONSTRATION MODEL.

low speed is required. Several materials were considered for the static rubbing ring; it was felt that an oil filled porous metallurgical product offered the best features. In all material considerations the close proximity of mercury was not forgotten. Thus in looking ahead, a non-amalgamating oilite, say a nickel material, may be suitable for this element.

From the start it seemed obvious that the only suitable collar material was rubber. Due to the probable presence of oil a neoprene was indicated; durometer hardnesses of 40 and 60 were thought to lie in the appropriate range. It was also believed that slaked graphite could be applied to the rubber compound prior to moulding; the precipitating action of the graphite on the rubbing surface lessens the heat generated.

The action of the coiled garter spring is rather unique. Light springs of this form have been used on "snap-off" seals; they have been ballasted with steel balls for immediate breakaway upon rotation. In our problem the necessity for delay has been made clear. Thus, prediction becomes important. In this design the assumption was made that all the *elasticity* of the system resided in the spring with extremely small interaction in the rubber-to-spring attachment. No separate ballast was considered; calculations presumed the centrifugally affected mass to be made up largely of the spring itself. This seemed reasonable since the delay obviously required quite a stiff spring. Thus in *prebreakaway* operation the running force on the spring at its axis would be made up of: (a) centrifugal force, (b) the interference pressure corrected to the spring axis, and (c) inward directed forces acting on the *effective width* presently to be discussed. The action of the spring was estimated on its approximate performance as an extension spring under the *net* force. The formulae used are shown in the appendix.

The mechanism of low speed sealing action, seal-breakaway, and closure are rather complex. It was initially felt that breakaway would occur when the pressure of interference between the collar and the rubbing ring was equal to the ambient hydrostatic pressure. Thus a substantial interference must exist at no

speed; this could be accomplished by initial spring adjustment to a predetermined radius. An assumption had to be made as to the effective width of the rubber collar over which the hydrostatic force influenced the interference pressure. This is a direct addition to the interference pressure. The cycle of operation under pressure is shown in the appendix where the quantity p_{es} is the *total* initial interference pressure. It was estimated that the action of breakaway would be one of instability since the addition of water under pressure on the inner side of the rubber collar actually is tantamount to a direct outward load addition. Thus a rapid outward collar displacement would take place. This also explains, by similar reasoning, as shown, the lower speed of closure when the shaft slows down. This reasoning allows the preparation of appropriate curves of breakaway and closure. Characteristic curves, Fig. 11 are shown in the Appendix. These curves are empirical to the extent that the breakaway pressure ratio p_{eb}/p_w, was taken equal to .552. This was based on experiment with an assumed effective width of collar. It was evidently constant for all pressures.

C. CHARACTERISTICS OF OPERATION

As discussed above, the characteristic curves are in part made up of anticipated breakaway and closure speeds for various pressure conditions that are functions of the low speed seal. Implementing these, on Fig. 11, are curves based on minimum seal of mercury for various speed lags. These curves take conservative account of the increment centrifugal water pressure above the true hydrostatic value. The theoretical range of a given design, then, is the pressure corresponding to the intersection of closure speed and the minimum mercury speed for the correct lag in mercury velocity. Actually an overlap is logically called for in safe design. Closure should take place at a speed in excess of the minimum mercury sealing speed. This is the seal margin of safety. The final adjustment of this can be made by regulation of the spring tension.

IV. Tests

A. PROTOTYPE DESIGN

Fig. 3 and certain of the curves in the appendix pertain to a demonstration model which was built by Reed Research, Inc. The purpose of this model was to demonstrate the feasibility of a mercury seal incorporating the features discussed above. A secondary aim of the work was to predict performance for a large high speed shaft.

It was felt that *one way* sealing action only need be demonstrated. Accordingly, the inside of the box, Fig. 3, represents a compartment under hydrostatic pressure against which the seal acts—from the inside out. Thus an overhung shaft was feasible. It would then be possible in later tests to demonstrate operation under simulated conditions of misalignment. It seemed desirable to completely separate the mercury seal and the low speed elements. A more or less cylindrical form of rubber collar was selected. A number of collars were actually built from the same mould. These varied in hardness and in slaked graphite content. Although segmented springs and snaprings were considered, the garter springs seemed to possess outstanding advantages.

The static ring was designed without lubrication grooves. It was assembled with a press fit to the supporting flange

and held in place by means of a locking nut, Fig. 3.

For assembly purposes the static disk had to be separated from the flange. The geometry of the shells as shown follows the necessity of retaining the mercury when the device is at rest, and of avoiding splash when starting from rest. It also reflects a desire for flexibility and more liberal manufacturing tolerances. At top speed the mercury pressure at its outer circumference is quite high. The gasket therefore required considerable attention. A high rubber-content cork material was selected. The retaining shell was deemed necessary as insurance against dynamic instability of the springs and collar at maximum speeds. The rubber collar was designed with 0.125 in. interference on the inner shell. It was secured with a "tension cement." This was desirable on the model, although for standard installation a vulcanized assembly would probably be advisable.

As noted on the accompanying photographs, Figs. 4-7, speed variation was accomplished through a double variable pitch control. A speed range of from 500 to 4000 rpm was thus available.

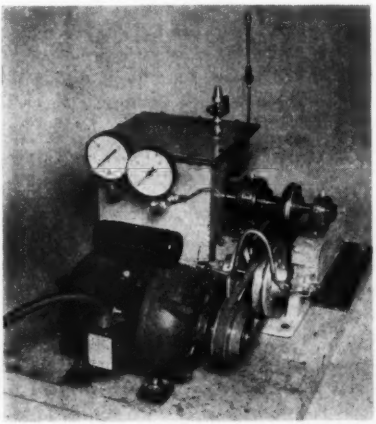


Figure 4.—Overall View of Demonstration Model—Showing Motor, Variable Speed Control, Pressurized Compartment, Forced Lubrication Line, and Pressure Gauges.

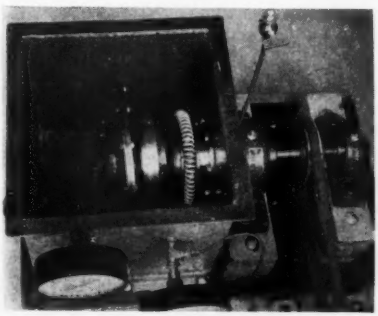


Figure 5.—Demonstration Model Seal—Showing External Components; Retaining Shell has been removed to show Rubber Collar and Garter Spring.

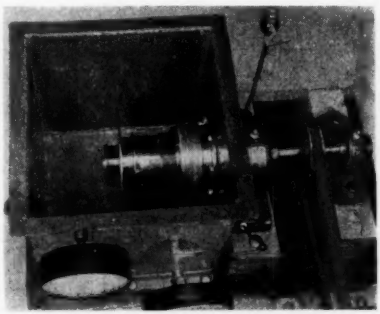


Figure 6.—Model Seal—Static Assembly—Showing Static Disk, Rubbing Ring, and Compartment Flange.

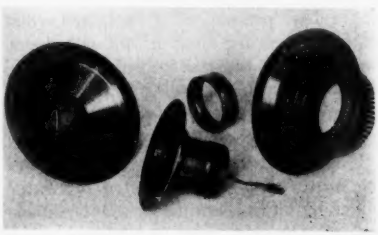


Figure 7.—Model Seal Elements—From left to right: Outer Shell with Gasket, Static Disk, Rubbing Ring (showing oil grooves), and Inner Shell with Collar and Spring Assembled.

B. RESULTS OF TESTING

Both seal components were tested separately under air pressures up to 12 psi prior to full testing of the assembled unit. An oilite rubbing ring, filled with SAE 10 oil was used. Breakaway occurred at about 1400 rpm which was quite close to the predicted breakaway.

The testing of the mercury action alone, aside from confirming the general theory, served the important purpose of evaluating the speed for minimum mercury seal. A general idea of mercury action at lower speeds was also obtained. When the mercury seal failed, the air, or water, did not displace the mercury from the outer shell region but appeared to slip inside and around the edge of the static disk. It would seem that the mercury, under this condition is flattened against the two shell halves with an annular escape passage for the fluid inside. These early tests tended to confirm the design estimate of a lag of about 35 per cent between the mercury and the shell.

The two seal components were then tested as a unit under air pressure. Then the housing was charged with water to a pressure of 10 psi. In early testing the first test of a series was entirely satisfactory; subsequent tests of a series brought to light a stalling tendency at the breakaway speed. Even with a ½ hp motor driving the unit breakaway could not be repeatedly effected. This condition was particularly pronounced at higher pressures. A polished steel rubbing ring and then an oilless powder metallurgical brass were substituted for the oilite ring. The same high pressure stalling was observed and was even more pronounced.

The conclusion was that oil flow from the oilite pores was inhibited by water under high head, and consequently lubrication after the first breakaway, of a test run or series, was greatly impaired. The answer seemed to be to arrange for forced lubrication of the oilite ring for high pressure operation. Therefore, another oilite ring was machined as shown in Fig. 3. Internal oil grooves were machined and oil admitted to the inner surface at four points. Subsequent tests proved that for operation under pressure of water a lubricant supply at higher pressure permitted trouble-free action.

It will be noted on the analytical characteristic curves of Fig. 11 that closure speed becomes increasingly lower with added head. This was noted in the model. In fact it was suspected that the centrifugal action of water immediately inside the rubber collar on slow-down tended further to inhibit closure. This action will be discussed in detail in the following paragraphs.

The device was tested under conditions of slight shaft-bulkhead misalignment. While the action of the seal elements did not seem far different from aligned tests, these results are inconclusive. This was not felt to be a part of the first phase testing.

Gasketing of the mercury container (i.e. the shells) does not present a serious problem. However, it was found that considerable care should go into the scarfing of the ends of the gasket. Evidence of a light mercury spray on the inside of the oilite rubbing ring was occasionally found. It is of sufficient importance to suggest the inclusion of internal splash guards.

The spring was wired to the rubber collar and no signs of tearing or pulling were noted. The ends of the spring were made fast, one to the other, by means of a short length of metal tubing. At present, adjustment is made by varying the effective length of the spring. With the spring as adjusted for test the limiting water head of the model was about 7 psi.

Temperatures do not appear to present any difficulties. Even under early test stalling, it is doubtful whether the ring or rubber temperatures at any time exceeded 150°F. The temperature gen-

erated at high speed between the mercury and the static disk was also moderate.

The repeated operation of the seal permits a small amount of fluid to be transferred from the high to the low pressure side of the seal. An inspection of Fig. 3 will show that water may enter the annular shell when the rubber ring lifts from contact with the static ring. Some of the fluid is then permitted to pass to the other side of the seal housing *only* when the shaft is brought to rest. The amount of water that would be transferred in this manner is of no consequence in a bulkhead seal. The quantity may be reduced to a negligible amount by a design that keeps the space between the shell walls and the disk at a minimum.

C. FUTURE TESTS

In essence, the present model presents a good tool for the further experimentation which seems warranted. The present collar shape is not, perhaps, the optimum configuration. Investigation of other collar shapes is desirable. A disk form of rubber element appears to offer advantage. It would compact the seal; and any centrifugal excess pressure, acting from the inside, would minimize the delay in closure of the seal on slowdown. A number of other factors should be considered in any further test of this seal. These include the following:

1. *Effective width.* In order to predict the behavior of the spring, the effect of hydrostatic pressure is important. The extent to which this pressure influences the spring action should be learned. This might be accomplished by sealing off the collar under varying internal pressure as a separate component test and by measuring relative spring displacements (i.e. double collar-spring testing).
2. *Mercury distribution.* An electrical probing or sounding of mercury under various conditions of sealing or no sealing is desirable in order to investigate the fluid dynamic action.
3. *Lubrication.* Testing of new type powder metallurgy products for the rubbing ring would be of value. Instrumentation for frictional power losses should be made. Frictional drag of the mercury could be studied (i.e. torque measurements).
4. *Deformation.* The model, as it now exists, is equipped to test a small degree of misalignment. Testing under gross misalignment with expandable bellows and internal aligning rollers would be valuable. This would determine the exact degree of maximum satisfactory misalignment
5. *Spring action.* The unusual use of a garter spring suggests some quantitative measurements of the spring under the action of centrifugal forces.
6. *Mercury rotation.* The possible artificial roughening of the inner shell surface is interesting. This may reduce the lag between mercury and the rotating shell. A vaning of these surfaces might be considered.
7. *Geometry.* Narrowing of the mercury area may result in a more compact unit, especially with split shells. Greater differences in radii may be possible and would result in higher operating pressures for the same speed.

V. Possible Fields of Application

The proposed application of this mercury seal was for main shafting on ships. However, it is considered that this seal may also find use in compressors, pumps and other applications. Several possible uses are described briefly in the following paragraphs.

A. LINE SHAFTING

The sealing of line shafting at points of bulkhead penetration appears to be an ideal application of the mercury seal with features incorporated to give maximum flexibility. This flexibility is necessary to assure sealing under severe bulkhead damage and one compartment flooding that might result from a collision between two vessels. The components of a basic seal, as shown in Fig. 8, could not handle severe misalignment. The attachment of the static rubbing ring to a deformable bellows, Fig. 8 B, is indicated as a means for obtaining high flexibility. In order that all deformation take place in the bellows, it is necessary that the static elements of the seal proper remain in reasonable alignment with the rotating units. The alignment is possible to retain by utilizing circular rollers and races in the annular channel and the fixed radial disk as shown.

Bulkhead distortions, depending on shaft size and seal dimensions, as great as 12 inches are considered feasible without causing the seal to become inoperative.

It is entirely feasible to operate under normal conditions with the seal dry of mercury. Provisions for automatic mercury fill with the rise of water to a predetermined level are practical. Therefore at high speeds, under normal conditions, the losses in the seal would be negligible.

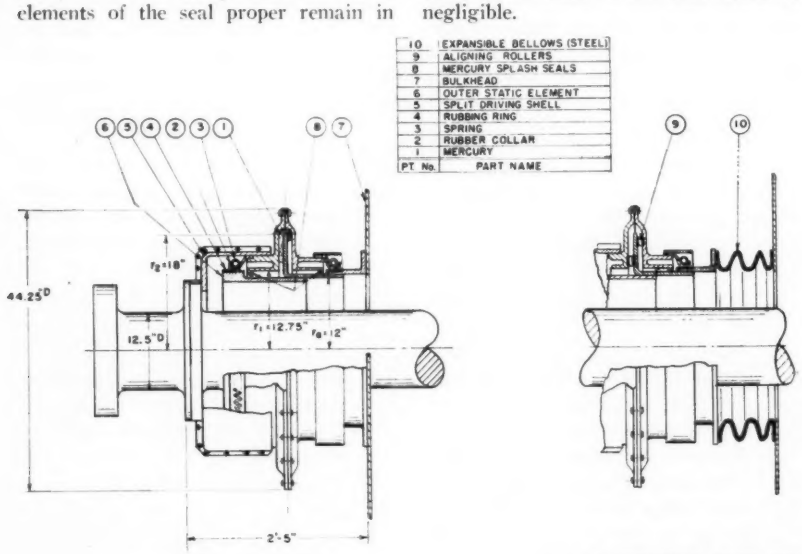


FIG. 8 SAMPLE LINE SHAFT APPLICATION.
MAX. SPEED ~ APPROX. 350 RPM
EST. BREAK AWAY ~ 130 RPM AS SHOWN

B. GEAR CASES

The problem of sealing a shaft at the point of entry to a gear case is somewhat simplified over that needed for a bulkhead as described above. The severe misalignment conditions between the case and shaft need not be considered because any appreciable misalignment at this point would render the gear inoperative. A seal as shown in Fig. 8 A would satisfy the conditions to be encountered under normal operation with small misalignment. With moderate heads, approximately 15 psi, to seal against, the proportions of the unit may be kept within the size indicated.

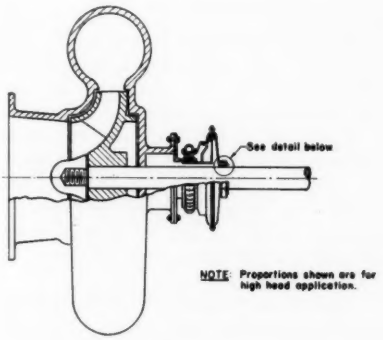
The low speed operation, where the flexible elements are rubbing the static ring, is in a range that is well below the normal operation and would be encountered only as a transient condition. The friction losses under these conditions would not be severe.

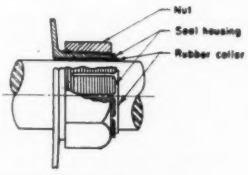
C. CENTRIFUGAL PUMPS

Centrifugal pumps seem to offer a particularly promising field for mercury shaft seals. Usually being constant speed machines, the low speed component could snap off at the start on such applications. The speed of pumps is usually fairly high and therefore high head sealing is quite possible. A mercury seal, correctly designed, is positive and absolute. Wear problems are negligible, and maintenance would not be a factor. Such an application is illustrated on Fig. 9. This application could be applied to existing pumps without necessity of shafting or housing modification.

FIG. NO. 9

SCHEMATIC APPLICATION OF MERCURY SHAFT SEAL TO CENTRIFUGAL PUMP





POSSIBLE ATTACHMENT TO EXISTING SHAFT

VI. Design and Manufacture

The design and testing of this seal has indicated a number of features that should be considered for inclusion in a full size unit.

One of the obvious factors is the proper selection of material. Any tendency for the formation of an amalgam must be avoided. It is obvious that aluminum or bronze will not be suited for the radial disk or the shells. Steel is considered satisfactory for these elements. However, aluminum and other light metals such as titanium may be used if they are given a protective coating of rubber, plastic or other material that will not amalgamate with mercury.

The flexible elements should be oil-resisting material such as neoprene, to prevent deterioration due to contact with a lubricant. Material selected for the rubbing portion of the static ring should be of a type that is self-lubricating for non-pressure operation (i.e. powder metallurgy product). Provisions for automatic lubrication under conditions of high pressure, say a flooded state, can be easily achieved. Constant speed units require no forced lubrication.

The design of the seal is such that two shell pieces are necessary. Due to pressures generated within the mercury charge during high speeds, use of gaskets between shells is essential to retain the sealing medium. The radial disk and the static ring may be integral units or made separate and secured by several methods. The garter spring has been selected as the most suitable type for the retaining unit since it is possible to predict with reasonable accuracy the action under varying speeds. Provisions can be made to mould the spring with the flexible element after exact spring characteristics have been determined. Due care must be exercised in the design of the shells to assure retention of the mercury charge that has been distributed around the circumference.

While actual test experience has not been obtained with flexible collars that were wrapped around the shaft and bonded to form a ring, it is questioned whether a flexible unit formed in this manner would be satisfactory. It would appear that a collar formed by moulding a continuous ring would give a better contact surface and would result in trouble-free operation.

The design of the seal should be such that the mercury charge may be withdrawn and replaced with a minimum amount of difficulty.

VII. Conclusions

The tests conducted on the model seal, Figs. 3-7, have indicated the basic feasibility of the mercury shaft seal for the application specified earlier in the paper.

Operation of this seal indicated that a system of forced lubrication is needed between the flexible element and the static ring *when hydrostatic pressure is applied.* This appears to be true even though a self-lubricating material is used for the static ring.

The delayed action of a spring-loaded flexible element can be attained through centrifugal action within certain limits.

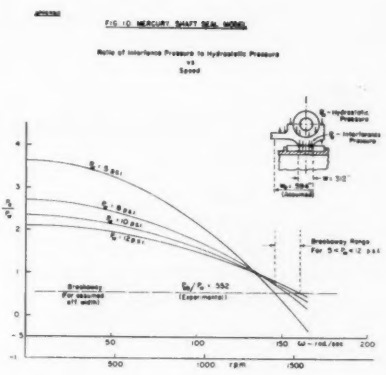
The flexible elements should be made as short, axially, as practicable. It is believed that this may result in a more compact design and will also enable the low speed seal action to be predicted with greater accuracy as to closure.

This mercury shaft seal, as at present conceived, of course, implies elaborations over present seal designs. The extra length and radii of the seal, the requirement of mercury, the greater engineering, and the number of components of different designs all leading to higher costs are superficially unattractive features. There is sometimes a feeling that a loss of simplicity is a loss of reliability. Objective decision should relate such a device to its place in the sun. That place can loom rather large if it is to safeguard the actual existence of any type of vessel. Penetration of a watertight bulkhead by a rotating shaft is obviously a weak link in the watertight integrity of a ship. The investment in extra insurance in the form of a more versatile seal is therefore small with regard to the overall investment. With regard to its reliability, the low speed and static components appear extremely reliable for all pressures.

At higher speeds an array of such seals in gear cases, along line shafts or in other application would absorb very little power. Line shaft seals operating dry would absorb *no* power. It is quite probable that line shaft seals could be

developed such that bulkheads may be moved considerably due to collision or other causes without loss of seal action even with operation over the entire speed range; shafts, themselves, could be moved appreciably without loss of seal action at points of penetration.

This seal makes two claims for itself which are believed not yet made by any other design. The first claim is one of improved functional performance as well as of power economy. The second claim is one of great flexibility and consequently one of greater overall reliability than any other present seal.



APPENDIX

$$\% \text{ LAG} = \frac{N_s - N_m}{N_s} \times 100$$

N_s = SHAFT SPEED

N_m = MERCURY SPEED

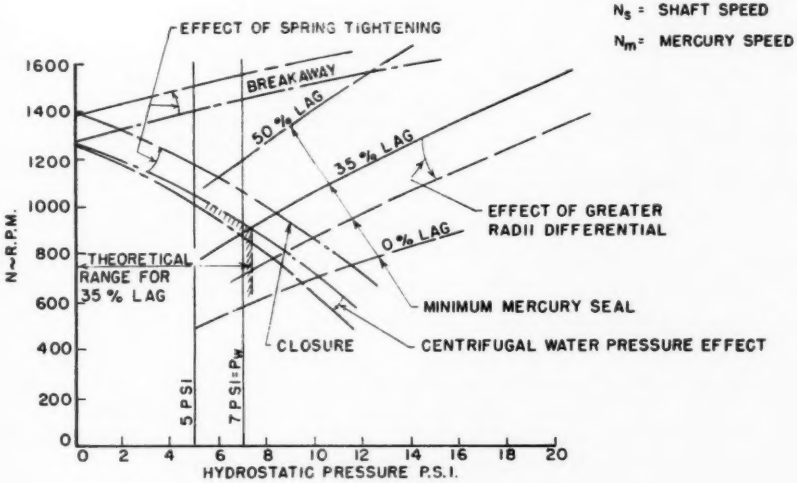
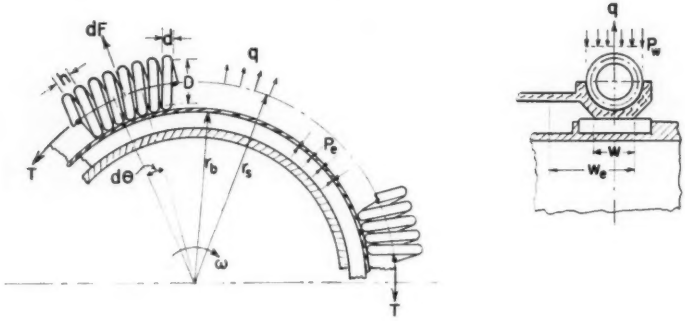


FIG II MERCURY SHAFT SEAL
CHARACTERISTIC CURVES DEMONSTRATION MODEL.

APPENDIX

APPROXIMATE GARTER SPRING FORMULAE



γ = st'l. density #/in³
N = no. of coils
F = centrf'l. force
T = spring tension
w = eff. seal width
ω = angular velocity — rads./sec
u = radial displacement
δ = spring extension — in.
w_s = spring wt. #/in
w_r = eff. rubber wt.
w_t = total rubbing wt. #/in = $w_s + w_r$

G = mod. of rigidity = 11.5×10^6 #/in²
h = mean spring pitch — in/coil
f = radius factor = r_s/r_b
S = stress in spring #/in²
P_e = rubbing pressure at radius r_b — #/in²
q = centrf'l. running load on spring at r_s — #/in
w_e = eff. width of hydrostatic action
P_w = hydostatic pressure #/in²
dF = elemental force on spring at r_s
dF_c = elemental centrf'l. force on spring at r_s

$$w_s \equiv \gamma \pi^2 D d^2 / 4h$$

$$N = 2\pi r_s / h$$

$$S \equiv T \cdot D \cdot 2.55 / d^3$$

$$\delta \equiv 8 \cdot T \cdot D^3 N / G \cdot d^4$$

$$u = \delta / 2\pi \equiv 1/2\pi \left(8 \cdot T \cdot D^3 N / G \cdot d^4 \right)$$

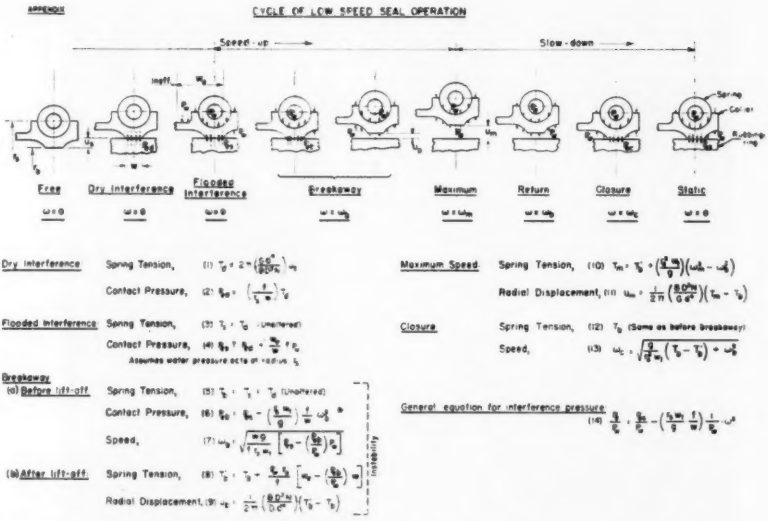
$$dF_c = r_s \cdot \omega^2 \cdot d w_t / g = r_s^2 \cdot \omega^2 w_t \, d\theta / g$$

$$q = dF_c / r_s \, d\theta = r_s \cdot \omega^2 \cdot w_t / g$$

$$q_t = q + \frac{1}{f} P_e w - \left(P_w \cdot w_e \right)^* \quad \text{- total running force on spring}$$

$$T = q_t \cdot r_s \quad \text{- net tension in spring}$$

* This term vanishes when hydrostatic pressure operates on inside of collar.


APPENDIX
CYCLE OF LOW SPEED SEAL OPERATION
Speed - up
Slow - down
Free
Dry Interference
Flooded Interference
Breakaway
Maximum
Return
Closure
Static
Spring
Collar
Rubbing ring
Dry Interference
Spring Tension, (1) $T_b = 2\pi\left(\frac{GS^3}{3D^3n}\right)w_b$
Contact Pressure, (2) $S_{bb} = \left(\frac{1}{t_b w}\right)T_b$
Flooded Interference
Spring Tension, (3) $T_b = T_b$ (unaltered)
Contact Pressure, (4) $S_b = S_{bb} = \frac{w_b}{w} S_c$
Assumes water pressure acts at radius r_b
Breakaway
(a) Before lift-off
Spring Tension, (5) $T_b = T_b = T_b$ (unaltered)
Contact Pressure, (6) $S_{bb} = S_{bc} - \left(\frac{r_b w_m}{g}\right)\frac{1}{w}\omega_b^2$ *
Speed, (7) $\omega_b = \sqrt{\frac{\pi \Theta}{r_b w_m}\left[S_{bc} - \left(\frac{r_b}{g}\right)S_c\right]}$
(b) After lift-off
Spring Tension, (8) $T_b = T_b = \frac{S_c r_b}{t}\left[w_b - \left(\frac{r_b}{g}\right)w\right]$
Radial Displacement, (9) $u_b = \frac{1}{2\pi}\left(\frac{8D^2N}{GS^2}\right)(T_b - T_b)$
Maximum Speed
Spring Tension, (10) $T_m = T_b + \left(\frac{r_b w_m}{g}\right)(\omega_m^2 - \omega_b^2)$
Radial Displacement, (11) $u_m = \frac{1}{2\pi}\left(\frac{8D^2N}{GS^2}\right)(T_m - T_b)$
Closure
Spring Tension, (12) T_b (Same as before breakaway)
Speed, (13) $\omega_c = \sqrt{\frac{g}{r_b w_m}(T_b - T_b) + \omega_b^2}$
General equation for interference pressure
(14) $\frac{S_b}{S_c} = \frac{S_{bc}}{S_c} - \left(\frac{r_b w_m}{g}\right)\frac{1}{w}\omega^2$
Integrity

* If an effective width is assumed, test data from one test will give a value of $\frac{S_{bc}}{S_c}$ which can be presumed constant for all pressures. This allows prediction of breakaway and closure.

VIII. Acknowledgments

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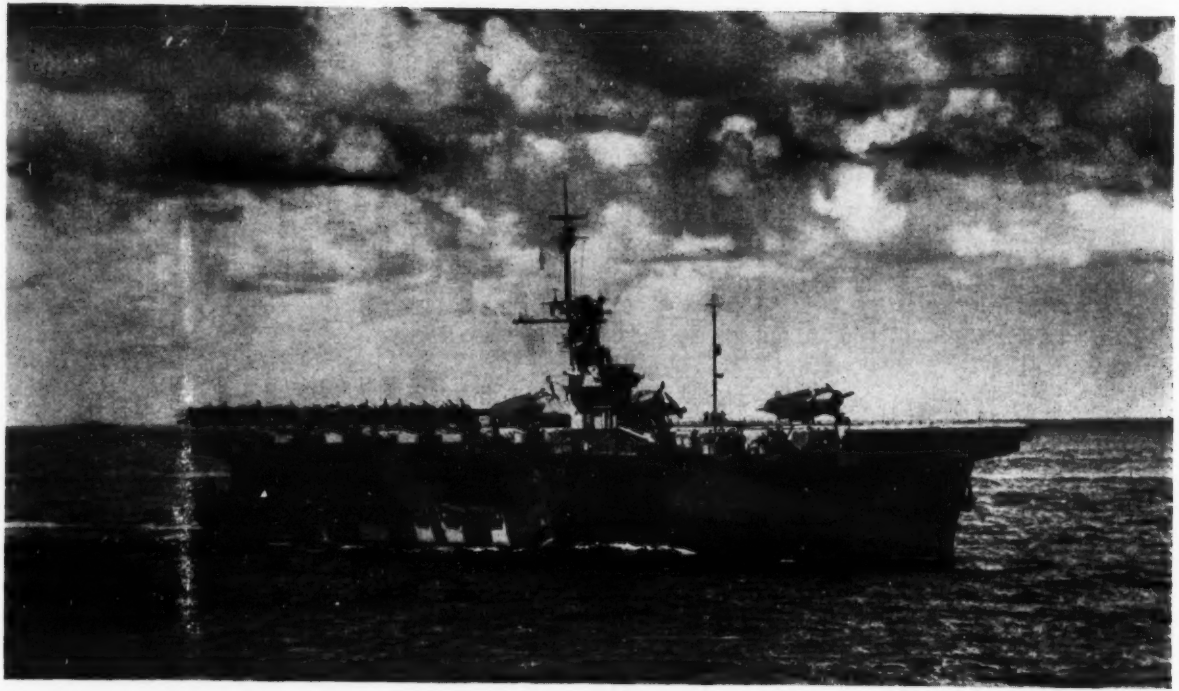
Keystone Carbon Co., Inc., St. Mary's, Pa.
Hunter Spring Co., Lansdale, Pa.



Official U. S. Navy Photograph

U. S. Wake Island C.V.E. 65

EXPERIMENTALLY DETERMINED CHARACTERISTICS OF CYLINDRICAL SLEEVE ANTENNAS

O. NORGORDEN

AND

A. W. WALTERS

THE AUTHORS

O. NORGORDEN (B-EE- '31 PhD '36). Received his B-E.E. and PhD. degrees from the University of Minnesota. He has been at the Naval Research Laboratory since 1936 and is now in the Communication Branch of Radio Division II. Present activities are concerned primarily with Communication System analysis, wave propagation and radiating systems. He has been active on the Radiating Systems and Communication Panels of the Research and Development Board.

A. W. WALTERS received a B.S. in Engineering at the University of South Carolina in 1938. Two and one half years general engineering experience in several plants in South Carolina. 1941 to 1942 as engineering inspector for the Inspector of Naval Material at Cincinnati, Ohio. From 1942 to 1945, active duty in the U. S. Navy. From 1942 to 1943 in the Navy's Radio Engineering course at M. I. T., Bowdoin College and Bell Telephone Laboratories. From 1944-45 as an Instructor in Navy Electronic training schools. Since February 1946 employed by the Communication Branch of the Naval Research Laboratory. The work is chiefly concerned with research and development on antennas and associated components, and in general, with application to shipboard communication problems.

Summary: A series of measurements of the impedance characteristics of cylindrical sleeve antennas has been made. These measurements were conducted over a wide range of frequencies, radiator and sleeve diameters, and ratios of radiator length to sleeve length. The data are plotted in families of curves of resistance and reactance vs frequency, considering each of the physical variables. These data may be employed in determining the physical dimensions of an antenna from the specific antenna impedance and frequency range specifications, or conversely in determining the electrical characteristics of a given physical design.

Measurement of the relative field patterns in a vertical plane are shown for representative antennas. Particular attention was given to this variation as a function of the ratio of radiator to sleeve length at a given frequency.

CYLINDRICAL SLEEVE ANTENNAS

I. INTRODUCTION

Some months ago the writers became interested in investigating the characteristics of Cylindrical Sleeve Antennas. This type of antenna is composed of elements, each of which have uniform cross sectional area, but are not necessarily of the same length or diameter. It can be considered electrically as a cylindrical monopole antenna with the feed point removed some distance from the base of the antenna.

This sleeve antenna does not necessarily constitute the ideal type for many designs, particularly when the diameter of the radiating element is large with respect to the feed line. Under these circumstances some advantage is gained by the use of tapered sections which avoid large discontinuities at the feed point and thus simplify the impedance matching problem.

It has been generally considered that, given the same physical cross sectional area, antennas of the "sleeve" type present a more desirable impedance characteristic to the terminating networks than other broad band antennas. A search of the literature has revealed reports on the design of various specific sleeve antennas,[1, 2] but has failed to reveal sufficient correlated data to provide general design information.

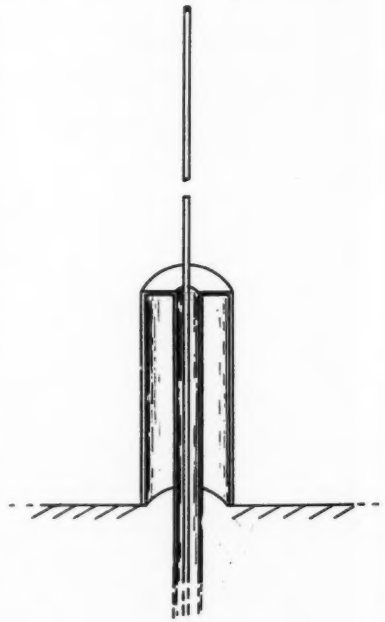


Fig. 1 - Sleeve Antenna

After considering various aspects of the sleeve antenna problem, a systematic experimental investigation was chosen in preference to a theoretical approach. The impedance data were obtained over a reasonably wide range of values for the four independent variables. The purpose of this article is to present these data in such form that the designer can apply it to a particular need.

II. PROBLEM DETAILS

The physical arrangement of the impedance measuring equipment is shown in Figure 2. A typical sleeve antenna is shown mounted on the ground screen in Figure 3. The ground screen was constructed of small mesh wire, with a brass plate in the center where the antenna was mounted, and was terminated on all sides by adjacent triangular wire mesh sections connected to a common ground strip. This type of termination greatly reduced reflections from the edge

[1] N. Lindenblad, RCA Review 12/4/39 "Television Antenna design."
[2] Bock, Nelson, Dorne; VHF Technique RRL Vol. I, Chapter V, "Sleeve Antennas."

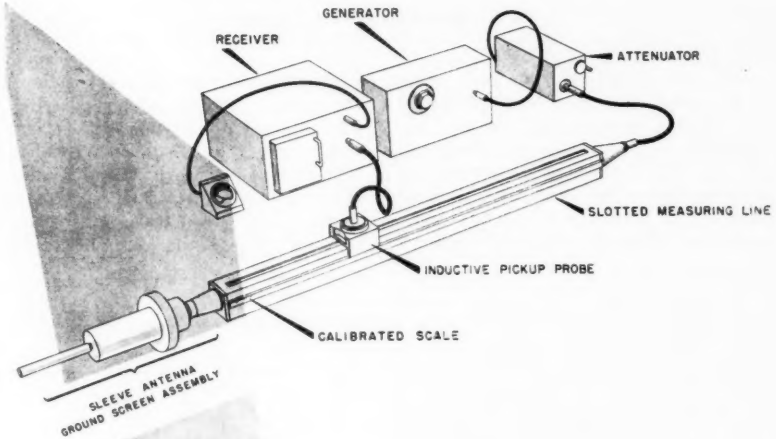


Fig. 2 - Sketch of Arrangement Used for Making Impedance Measurements

of screen and thus minimized the size of screen required. The ground screen structure was mounted on the side of building with the center at a second floor level since this permitted the use of an unusually short straight section of cable between the slotted measuring line and the antenna terminals.

Measurements were made by the normal probe-slotted-line methods with the system as shown in Figure 2. The probe-detector indicated the ratio of current maximum-to-minimum as well as the position of the current minimum on the line. These quantities, together with the position of the current minimum when the line is shorted at the antenna terminals and the impedance of the line, determine the impedance which exists at the antenna terminals.

It was found after considerable investigation that the optimum performance of the system shown in Figure 2 was between 65 and 330 Mcs. Then with 30" chosen as the fixed overall antenna length, the diameter of the upper *or radiator* section was varied from ¼" to 8", and the diameter of the lower *or sleeve* section was varied from 1 to 8 inches. The steps in each instance were in ratios of two to one, i.e. ¼, ½, 1, 2, 4, and 8 inch diameters for the radiators, and 1, 2, 4, and 8 inch diameters for the sleeves. The spacing between the sleeve and radiator was ⅜ of an inch for 2" and 4" radiators and 3/16 inch for all others. The outside diameter of the outer and inner conductors of the transmission line feeding the antenna were 1 inch and .25 inches respectively. The details of construction are shown in Figure 1.

Fig. 3 - Sleeve Antenna on Ground Screen

III. EXPERIMENTAL RESULTS

(A) Arrangement of the Data

The data resulting from each series of measurements have been plotted in two sets of curves, one showing the variation of resistance with frequency for the fixed physical dimensions of each member of the series, and the other showing the variation of reactance with frequency. Thus the curves of Figure 4 show the resistance characteristic for the series of sleeve antennas having a sleeve diameter of 1 inch, ratio of radiator to sleeve length of 4 and radiator diameters of ¼, ½, 1, 2, and 4 inches. This series has the smallest diameter sleeve considered in these measurments. Figure 5 shows the reactance curves

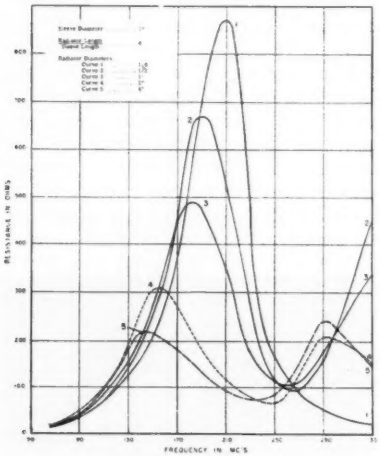


Fig. 4 - Resistance vs Frequency

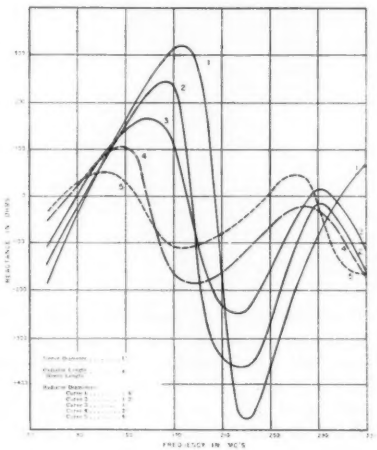


Fig. 5 - Reactance vs Frequency

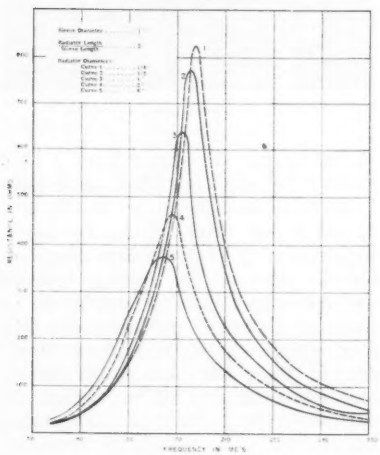


Fig. 6 - Resistance vs Frequency

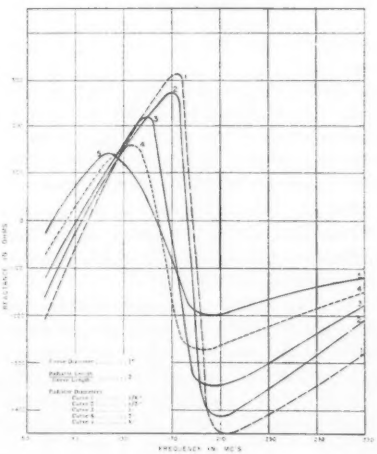


Fig. 7 - Reactance vs Frequency

for the same series of antennas. The next step in this family of curves for the sleeve diameter of 1 inch is shown on Figure 6. The radiator section is now twice as long as the sleeve, and the radiator diameters have again been varied from ¼ inch to 4 inches in 2 to 1 steps. Figure 7 shows the corresponding reactance curves. Similarly Figures 8 and 9 show the series with the radiator length equal to the sleeve length. Figures 10 and 11 show the series with the radiator length one half of the sleeve length, and Figures 12 and

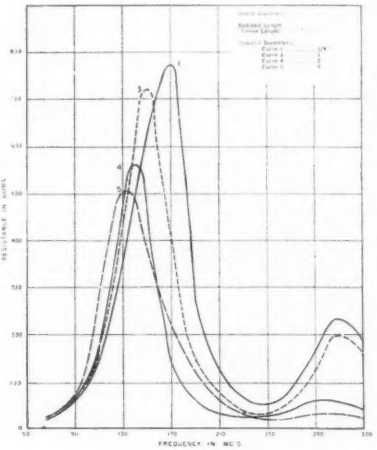


Fig. 8 - Resistance vs Frequency

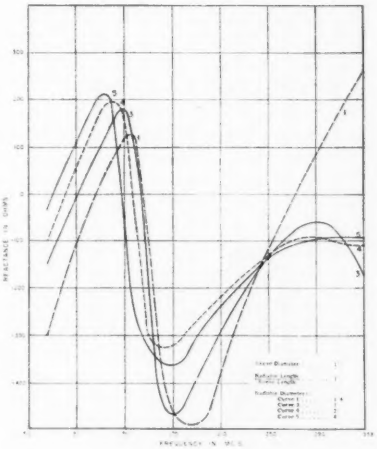


Fig. 9 - Reactance vs Frequency

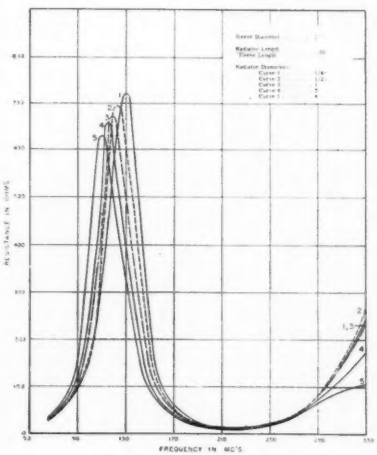


Fig. 10 - Resistance vs Frequency

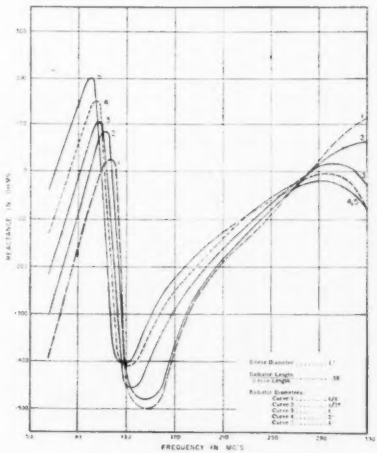


Fig. 11 - Reactance vs Frequency

13 show the series with radiator length one quarter of the sleeve length.

Figures 14 and 15 are the start of the family of data using a sleeve diameter of 2 inches. The sequence of the dimensional changes follows again as first described through Figures 22 and 23. Figures 24 through 33 present the data for the family using a sleeve diameter of 4 inches, and Figures 34 through 39 show the data for the family using a sleeve diameter of 8 inches. In general it can be noted that the highest value of maximum resistance occurs when the radiator diameter is smallest, and that the ratio of maximum to minimum resistance is largest for this same condition.

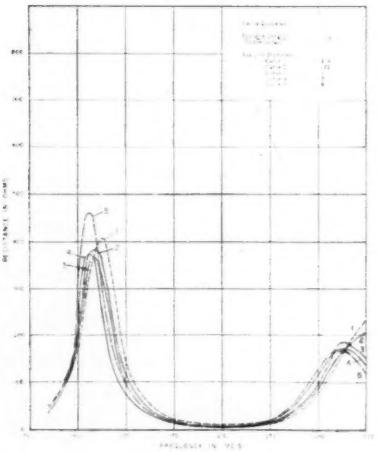


Fig. 12 - Resistance vs Frequency

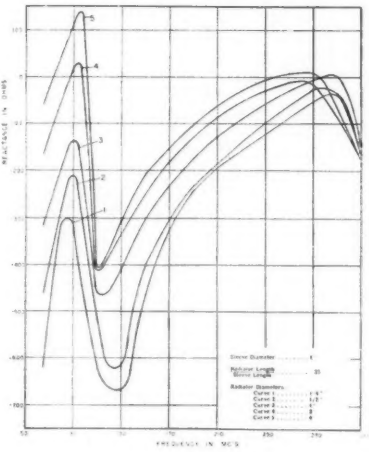


Fig. 13 - Reactance vs Frequency

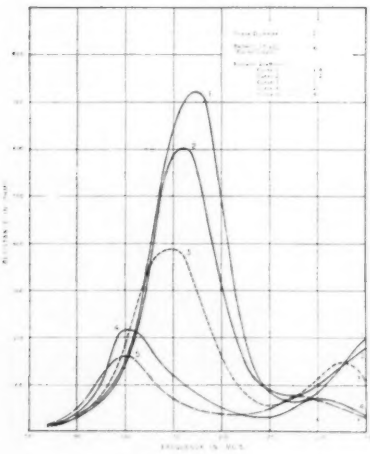


Fig. 14 - Resistance vs Frequency

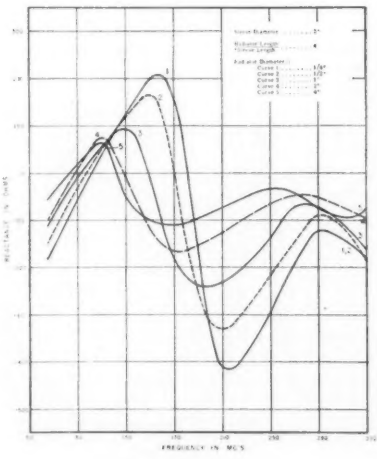


Fig. 15 - Reactance vs Frequency

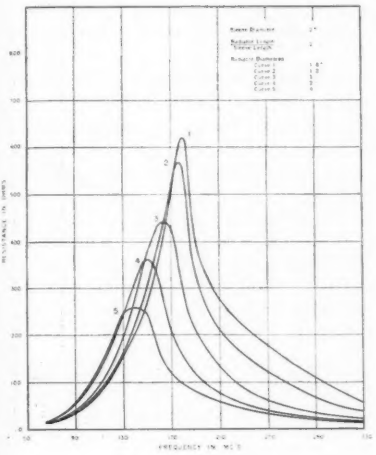


Fig. 16 - Resistance vs Frequency

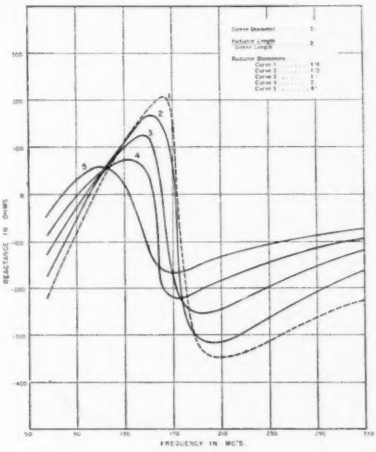


Fig. 17 - Reactance vs Frequency

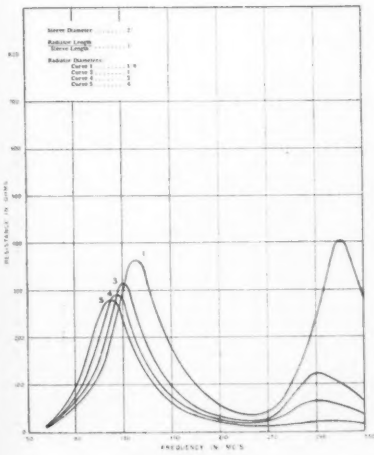


Fig. 18 - Resistance vs Frequency

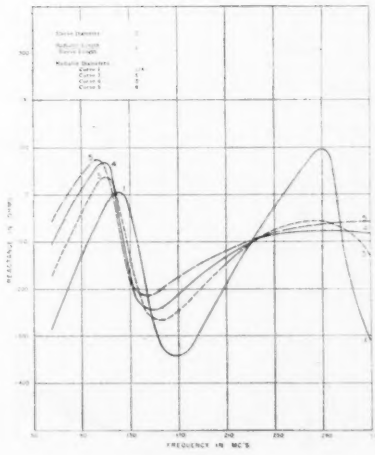


Fig. 19 - Reactance vs Frequency

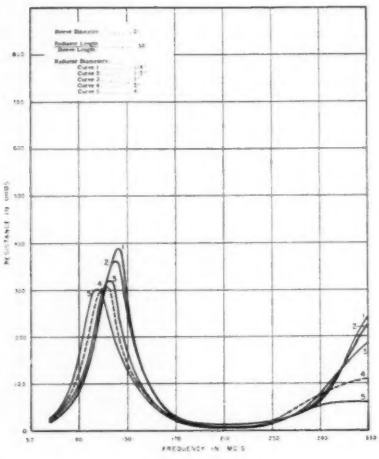


Fig. 20 - Resistance vs Frequency

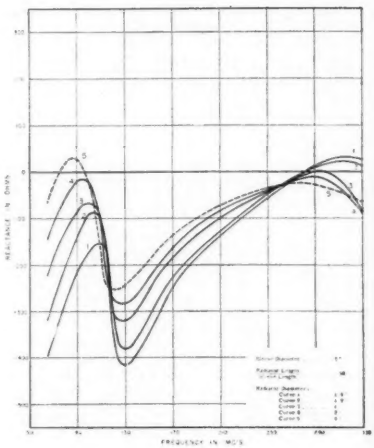


Fig. 21 - Reactance vs Frequency

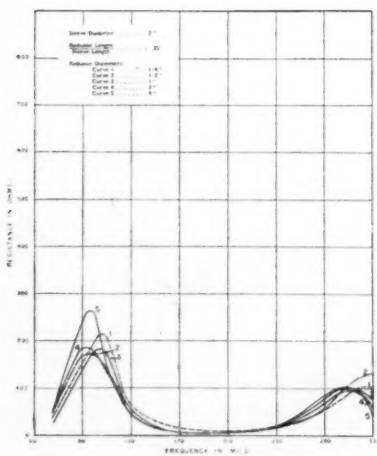


Fig. 22 - Resistance vs Frequency

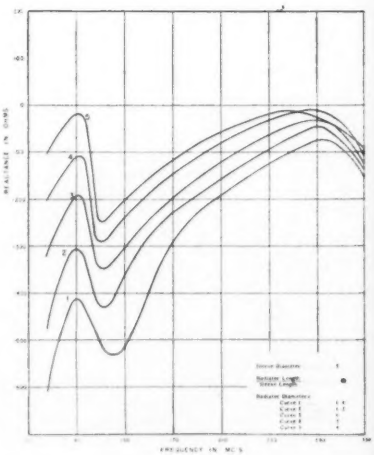


Fig. 23 - Reactance vs Frequency

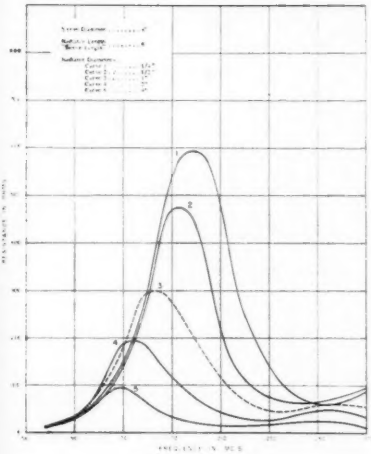


Fig. 24 - Resistance vs Frequency

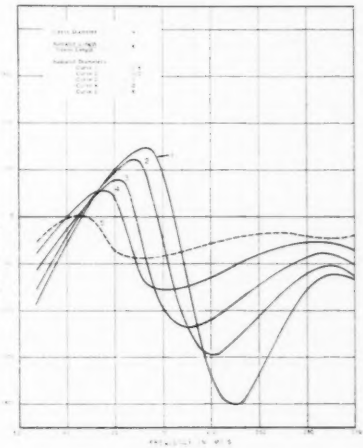


Fig. 25 - Reactance vs Frequency

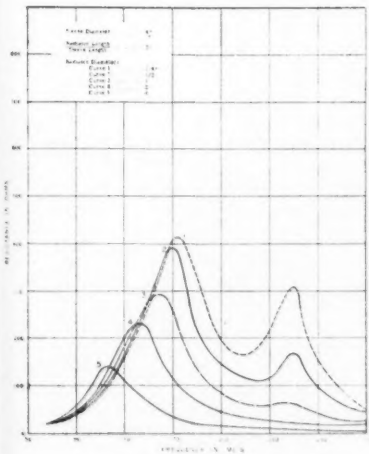


Fig. 26 - Resistance vs Frequency

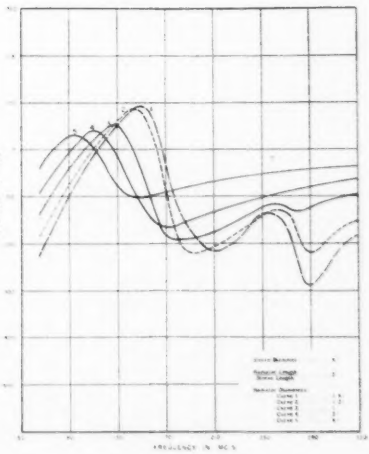


Fig. 27 - Reactance vs Frequency

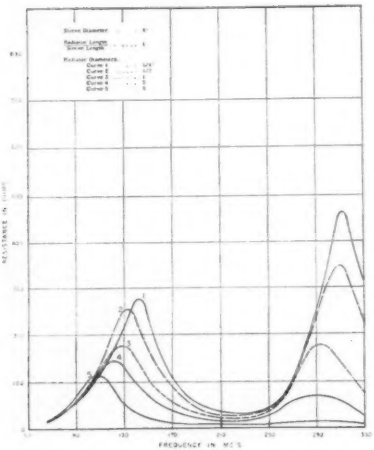


Fig. 28 - Resistance vs Frequency

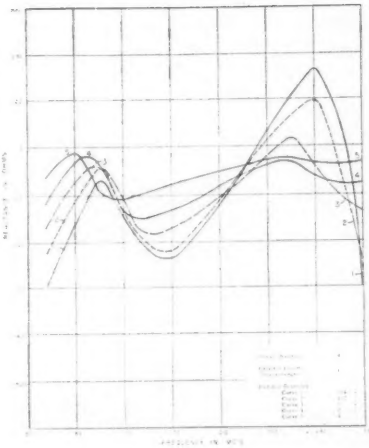


Fig. 29 - Reactance vs Frequency

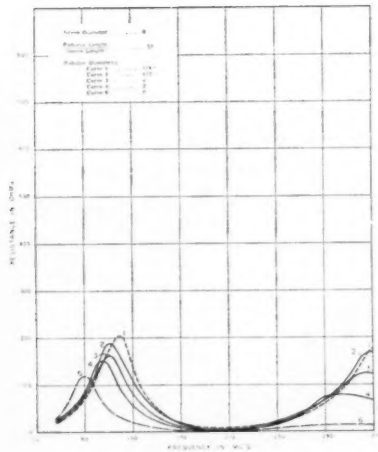


Fig. 30 - Resistance vs Frequency

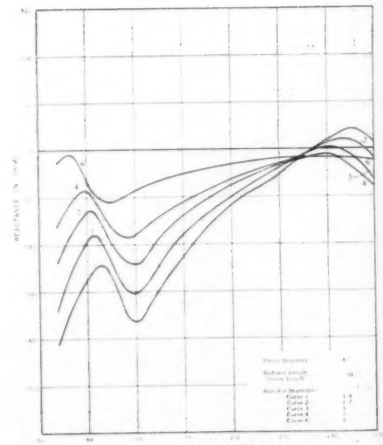


Fig. 31 - Reactance vs Frequency

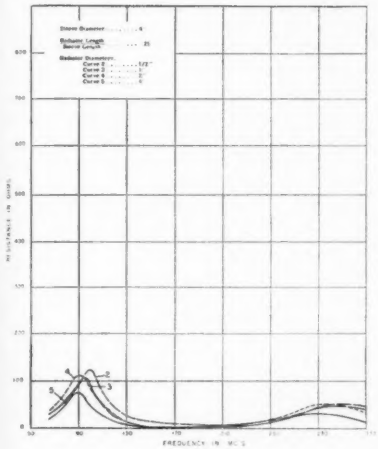


Fig. 32 - Resistance vs Frequency

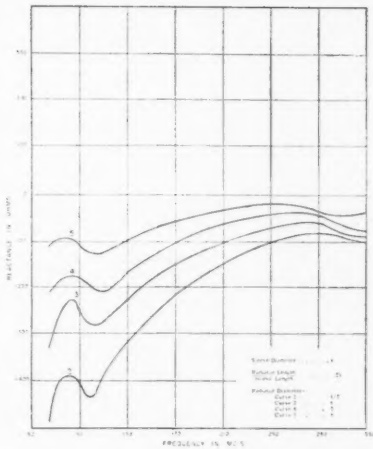


Fig. 33 - Reactance vs Frequency

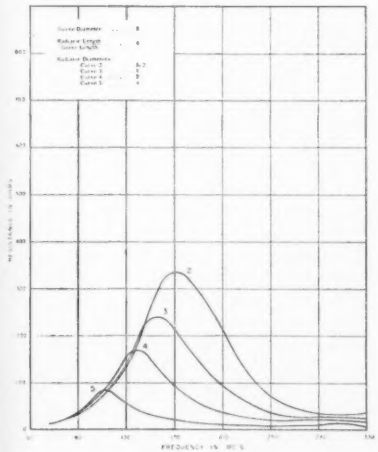


Fig. 34 - Resistance vs Frequency

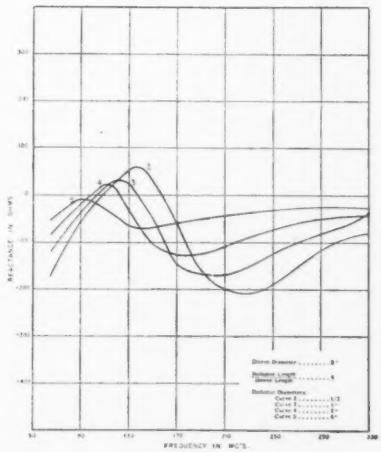


Fig. 35 - Reactance vs Frequency

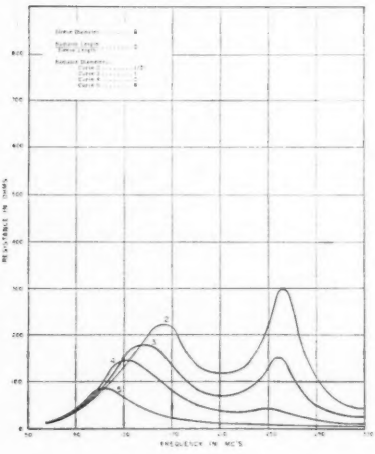


Fig. 36 - Resistance vs Frequency

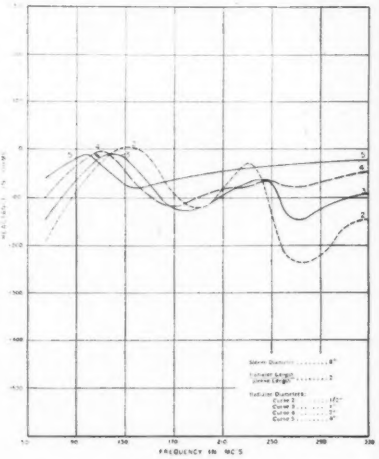


Fig. 37 - Reactance vs Frequency

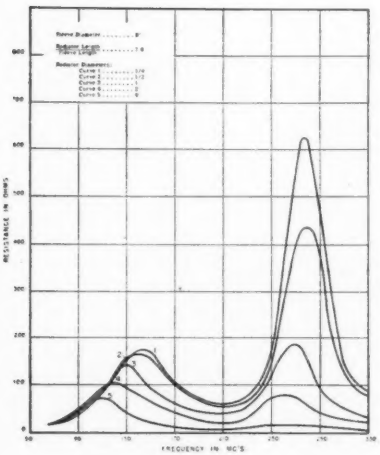


Fig. 38 - Resistance vs Frequency

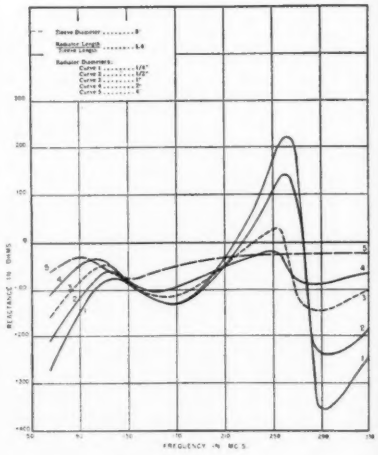


Fig. 39 - Reactance vs Frequency

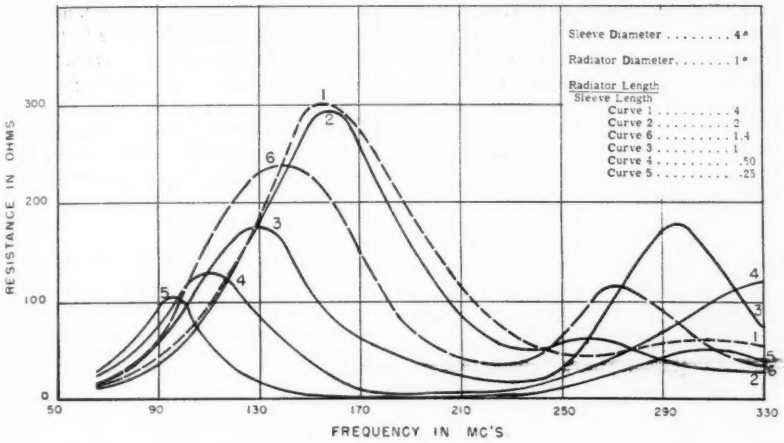


Fig. 40 - Resistance vs Frequency

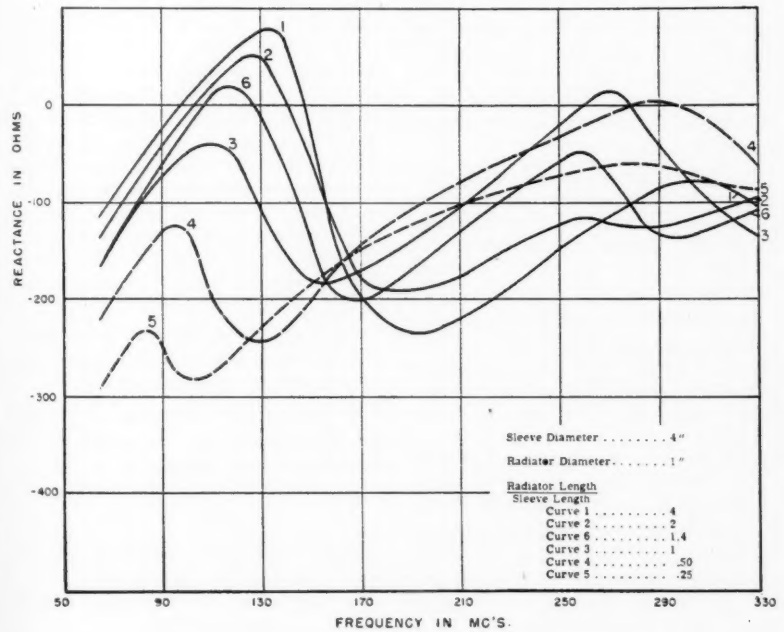


Fig. 41 - Reactance vs Frequency

The data presented in Figure 4 through 39 may be replotted in various ways considering two of the independent variables constant, and varying the remaining ones. An example of a different type of presentation is shown in Figures 40 and 41, where each curve represents a different ratio of radiator length to sleeve length for a fixed radiator diameter of 1" and sleeve diameter of 4 inches. It can be noted that in general, the position of the resistance maximum shifts towards the lower frequencies as the ratio of radiator length to sleeve length is decreased.

Application

The data as shown in Figures 4 through 41 are useful in the design of specific sleeve antennas for which the electrical or the physical characteristics have been predetermined. These designs may be classified as follows:

(a) When the physical parameters of a sleeve antenna have been restricted to specific limits, the problem is to design an antenna which has the most desirable impedance characteristic over a given frequency range;

(b) When the frequency range and standing wave ratio have been assigned limits, then the problem is to determine the physical dimensions which will produce an antenna having the desired characteristics.

As an instance where it is necessary to produce antenna to operate in a given frequency range with allowable values of standing wave ratio, consider the following case: Let the frequency limits be set at 65 and 300 Mc with values of standing wave ratios better than 0.20 to be obtained throughout. Then by considering the data of Figures 4 thru 39 it can be observed that the antennas represented by Figures 4 and 5 have reasonable impedance characteristics over the required band. The most promising of these consists of a 1 inch diameter sleeve, a 4 inch diameter radiator and a ratio of radiator length to sleeve length of 4. The resistance for this antenna varies from 17.5 to 245 ohms, the reactance varies from 50 to —140 ohms, and the standing wave ratio varies from .20 to .60. The impedance characteristics of this antenna could be considerably improved by the use of impedance matching sections.

Feed Point Capacitance

The physical construction of the sleeve antenna introduces a capacity that shunts the feed point. This capacity is in parallel with the true antenna impedance and cannot be measured directly. The two plates of this condenser for the sleeve antennas under consideration are the metal disc which closes the top of sleeve and the disc which closes the bottom of the radiator. (See Figure 1). The reactance of this shunting capacity is the same in any scaled frequency band for all antennas scaled from a given model. For a given antenna design this shunting capacity increases as the diameter of the radiator and sleeve increase. Hence the effect of this shunting capacity upon the antenna impedance is the greatest for the largest diameter radiator and sleeve. This fact is evident from the data because the antenna reactance is not symmetrically arranged about the zero axis but is shifted toward the negative reactance values. In many instances where both the radiator and sleeve are of relatively large diameter, it will be desirable to reduce this shunting capacity by proper design considerations.

By neglecting the fringing effect upon the capacitance, the following simple formula can be employed.

$$C = 0.02235 \frac{KA}{d} \text{ mmfd}$$

Where K—dielectric constant (air in this case and is one).

A—Area of the smaller disc in square inches.

d—spacing between the discs in inches.

Thus the shunting capacitance C is determined by the smaller diameter element (sleeve or radiator) and is essentially independent of the diameter of the larger element.

Since this capacitance is inherent in the construction of the sleeve antenna, no correction for its effect has been made in the foregoing data. However, its value has been calculated for two general conditions, namely a two inch diameter disc and for a 4 inch diameter disc with a spacing of ⅜ inches. Curves of the reactance of these two feed point capacitances are plotted on Figure 42.

An illustration of the effect of the

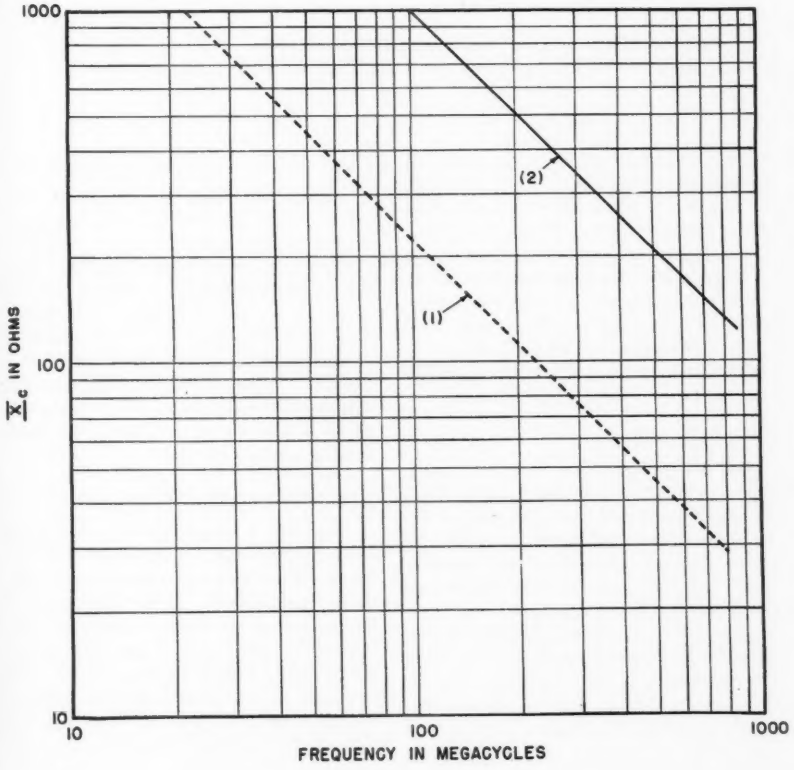
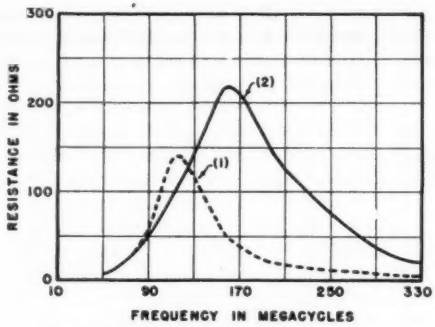


Fig. 42 - Shunt Reactance vs Frequency
Curve (1) - 4″ diameter disc
Curve (2) - 2″ diameter disc



Curve (1) . . Resistance vs Frequency
Curve (2) . . Resistance vs Frequency (with the base reactance effectively removed)

Fig. 43 - Sleeve Antenna, Resistance vs Frequency for 4" Diameter Sleeve, 4" Diameter Radiator and Radiator Length to Sleeve Length Ratio of 2.

feed point capacitance is provided by the curves of Figure 43. For these data the measured resistance characteristic of the antenna having a 4 inch radiator and a 4 inch sleeve (previously shown on Figure 26) has been taken point by point and corrected, giving the second curve on the Figure. It is seen that both the magnitude and position of the resistance are materially altered by the feed point or "base" capacitance.

Antenna Scaling

The model technique is a well established principle in antenna work. Thus the antenna impedance data are valid for other frequency bands providing all of the antenna dimensions are properly scaled. If for a given antenna design the frequency band has to be scaled by a factor "h", all of the physical dimensions of the antenna must be scaled by a factor of $1/_h$.

A simple check was made on the model principle by using a scaling factor of one half for all the physical dimensions of the antenna and then measuring the impedance at twice the original frequencies. The antenna impedance data for this scaled antenna were the same as for the original antenna.

Dipole Type of Sleeve Antenna

All of the data presented in this article have applied to the monopole type of sleeve antenna. If it is desired to have a balanced sleeve antenna; then the ground plane is replaced by the image of the sleeve antenna. The resistance and reactance of the dipole type of sleeve antenna are twice that for a ground sleeve antenna. The overall length of the dipole type of sleeve antenna is also twice that for a monopole type of sleeve antenna.

Antenna Patterns

A series of measurements of vertical field patterns has been made. It was found that field patterns are essentially independent of the diameters of the radiating elements within the problem limits previously defined. Considerable effect, however, was noted with changes in radiator-length to sleeve-length ratio. In general, patterns are similar to those of simple vertical antennas at frequencies for which the antenna is short with respect to a wavelength but as the frequency increases a decided null occurs in the region 20-30 degrees above the horizontal axis. The depth of this null is determined pri-

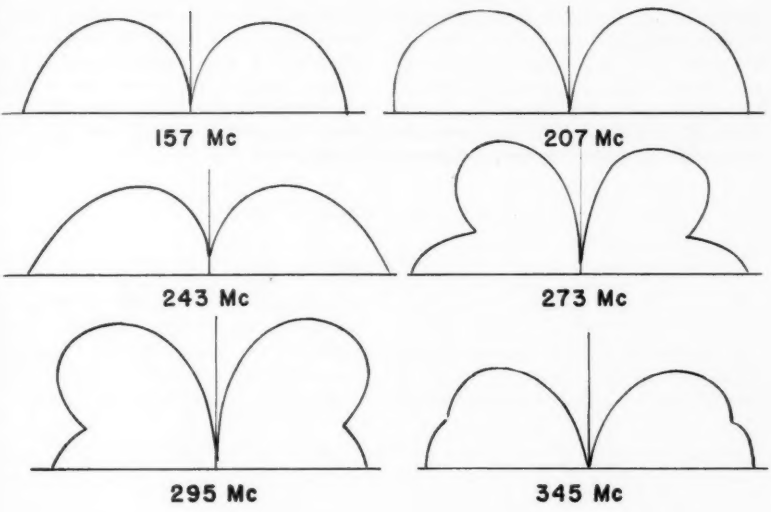


Fig. 44 - Vertical field patterns of sleeve antennas with fixed diameter radiator (1/2"), sleeve (4"), radiator length to sleeve length of 4, and a total length of 30" at frequencies as shown.

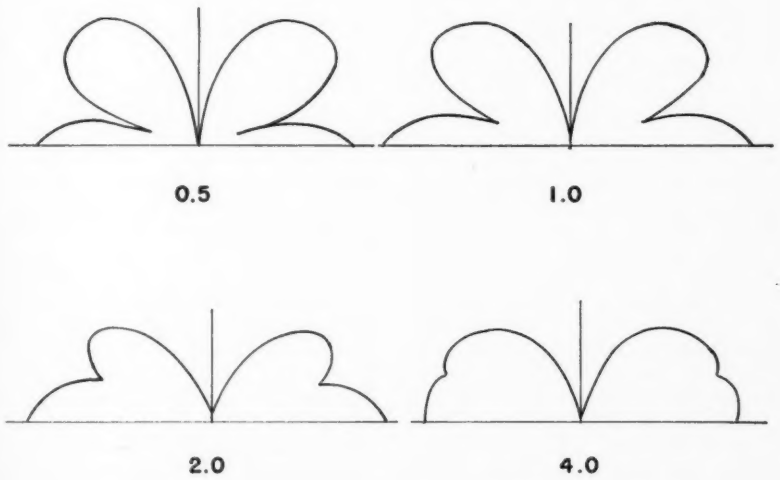


Fig. 45 - Vertical field patterns of sleeve antennas with fixed diameter radiator (1/2"), sleeve diameter (4"), and total length of 30" at 345 Mc. Ratios of radiator length to sleeve length are indicated under each Figure.

marily by the ratio of radiator-length to sleeve-length of the antenna, becoming deeper as this ratio is decreased. The patterns for a sleeve antenna in a vertical plane are shown in Figure 44 for 157, 207, 243, 295 and 345 Mc. This antenna has a ½ inch diameter radiator, 4 inch diameter sleeve and a ratio of radiator length to sleeve length of 4, and a total overall length of 30 inches. Figure 45 shows the vertical patterns for ratios of radiator-length to sleeve length of 4, 2, 1, and ½, at a frequency of 345 Mc.

CONCLUSIONS

Measured values of the resistance and reactance of cylindrical sleeve antennas, considering each of the independent parameters, have been presented in graphic form. The data presented in this manner should prove of value in the design of specific cylindrical sleeve antennas, or, conversely, as an aid in the prediction of the electrical characteristics of a given physical design.

A NEW NAVAL FIRE FIGHTERS' SUIT

DAVID F. RICHARDSON AND ROBERT GRUBB

THE AUTHORS

DAVID F. RICHARDSON received the degree of B.S. in Chemistry from the Worcester Polytechnic Institute in 1923, and an M.S. in Chemistry from George Washington University in 1925. From 1923 to early 1926 he was employed as a junior chemist at the National Bureau of Standards, initially in the lubricants and later in the lime section. From March to October of 1926 he conducted research projects for a commercial lime company. He was employed at the Industrial Test Laboratory of the Philadelphia Naval Shipyard as an assistant chemist in October, 1926, and is at present Superintendent of the Development Branch of the Laboratory.

ROBERT GRUBB was graduated "cum laude" from St. Joseph's College, Phila., Pa., in 1941 with the degree of B.S. in Chemistry. From June, 1941, he was employed in the Analytical Branch of the Industrial Test Laboratory, Philadelphia Naval Shipyard, to assist in the testing of leather, rubber and textile items. During 1943, he was transferred to the Development Branch of the Laboratory and is currently Head of the Protective Clothing and Impregnated Fabric Group.

INTRODUCTION

In fire fighting and damage control operations there is great need for a protective garment which will enable the wearer to enter flame- or steam-filled atmospheres to perform immediate rescue or salvage operations in advance of extinguishing the fire or repairing the damage by conventional means. The need is particularly acute in the Navy for many reasons. Stores of inflammable liquids and ammunition carried on board Naval vessels create special hazards even under normal operating conditions; in combat, fires involving such materials are frequent. Aircraft operations are hazardous enough on land; chances of crashes are multiplied when the landing field is the deck of a moving carrier. The use of steam at extremely high pressures has increased operational efficiencies manyfold, but has also increased the possibility of ruptured steam lines, and danger to personnel when breaks do occur. Add to these hazards the fact that space limitations aboard vessels make fire fighting and damage control operations especially difficult, and it is easy to visualize the value of a protective suit which will make it possible for the fire fighter to enter the emergency area without delay, even if for only a short period. Speed in closing steam or fuel-line valves, or in securing hatches, may represent the difference

between salvage and loss of a vessel; the value of speed in the rescue of personnel trapped in blazing aircraft or ships' compartments needs no emphasis.

The Navy has long recognized the usefulness of such a protective garment, and, as a means of partially satisfying the need, currently issues an asbestos suit to its fire fighters.[1] This suit was originally designed as an anti-flash garment and consists of a zippered coverall, incorporating gloves, shoes and helmet, made of a single thickness of asbestos fabric of relatively open weave. To provide strength and durability the cotton content of the asbestos fabric is high, and cotton thread is used in the fabrication of the garment. Because it is comparatively thin, and because of the thermal conductivity of the asbestos fiber,* the insulating value of the suit is limited and it will not provide more than momentary protection from flame. Its cotton content increases its vulnerability. Wetting the fabric increases the time required for the interior temperature to exceed safe limits, but introduces the very real danger of scalding the wearer by the steam generated when the wet suit is heated. Obviously, the suit is not safe for use in an atmosphere in which steam is already present. While wearing the suit it is possible for the fire fighter to use the standard Oxygen Breathing Apparatus,[3] and so equipped he may enter atmospheres deficient in oxygen, or in which toxic gases may be present. However, the O. B. A. must be worn outside the garment, and its combustible parts still further limit the exposure which the combination will withstand.

The limitations of the present suits are perhaps best indicated by certain statements and precautions which are included in official manuals. The Naval Manual on Structural Fire Fighting,[2] for example, states that: "Their use is objected to by some fire fighters, owing to the false sense of security which they engender. However, there are undoubtedly occasions when their use—with proper precautions—may be advantageous. A word of caution: Ordinary clothing worn under such suits will be ignited by the intense heat if the wearer remains more than a very brief period in flames. Moreover, when using the suit, it should either be kept dry or well soaked, as small amounts of water may cause steam burns." Similarly, the Bureau of Ships' Fire Fighting Manual [3] states, in part: "While asbestos will not burn, *it will char and conduct heat.* And therefore the asbestos suit affords protection against flames *for very brief periods* only. The length of time that the suit can be worn depends upon conditions under which it is used. Complete clothing should be worn under the suit to provide additional body protection. Furthermore, if the suit becomes wet, as would be more than likely in fire fighting, the wearer might be scalded, unless he withdrew from the heated area before the water turned to steam. Continued wetting would keep the wearer cool, but this procedure is seldom advisable, in view of the fact that the suit would become water-soaked, and reduce the wearer's freedom of movement, already restricted by the cumbersome suit. For these reasons, the uses of an asbestos suit in fire fighting are limited."

Emphasis on the limitation of the asbestos suit, and the thorough training of personnel in Fire Fighters' Schools, has been effective in preventing Naval casualties. However, several instances of serious injury to civilian fire fighters wearing the suits have been reported. In at least one case the fireman wore a suit into flaming gasoline floating on the surface of a pool of water in the course of a demonstration. As would be expected, the blazing liquid immediately penetrated the asbestos fabric, and the wearer received burns which were so severe as to endanger his life.

In spite of its deficiencies, the need is

* "Wool has only one-third the heat conductivity of asbestos." [3]

so great that the suit has become a most useful item of fire-fighting equipment. As the Bureau of Ships' Manual further states: "Notwithstanding its limitations, the asbestos suit is invaluable in certain situations. With it on, a fire fighter can move quickly through flames to effect a rescue, or for some other urgent task that can be accomplished quickly, one that would not be possible otherwise."

Recognizing the need, and the deficiencies of the best available equipment, the Bureau of Ships authorized the development of an improved fire fighters' suit at the Industrial Test Laboratory of the Philadelphia Naval Shipyard. Two major objectives for the project were stated: first, the development of an improved fabric or combination of fabrics to protect against heat and to be impermeable to water and other vapors; and, second, to design a garment which would make most effective use of the improved material. The most desirable fabric was defined as one which would, when made into a garment:

1. Maintain an interior temperature of less than 130° F. for the longest possible time when exposed to exterior temperatures approaching that of flaming gasoline;
2. be relatively comfortable; and
3. permit a fair degree of mobility to the wearer.

At the risk of being repetitious, it should be emphasized that it was in no way the purpose of the project to develop a garment which would replace the waterproof clothing ordinarily worn by the fire fighter while extinguishing fires, but, rather, one which would be used only in an emergency, permitting the performance of urgent tasks under the most extreme conditions of exposure.

PRELIMINARY STUDIES

As a first approach, the problem was studied by the Laboratory to define more precisely the exposure conditions prescribed by the Bureau of Ships, and to establish tentative requirements which a fabric, or combination of fabrics, should meet in order to be considered for the application. The Bureau had specified that the suit must be capable of withstanding "exterior temperatures approaching that of flaming gasoline." Temperature measurements were made in actual gasoline fires in simulated ships' compartments at the Fire Fighters' School of the Naval Damage Control Training Center. These were found to appoach 2000° F. in some instances; 2000° F., therefore, appeared to represent a reasonable maximum temperature which the suit would be required to withstand.

The companion requirement, that the suit should maintain an interior temperature of less than 130° F. for the longest possible time, is much less definite. It is immediately apparent that the time required for the interior temperature to reach this limit is a function of the insulating value of the suit material and of its thickness. The phrase "for the longest possible time" thus implies a compromise with the further requirement that the suit permit an acceptable degree of mobility to the wearer, since the insulating value is a constant for a given material and the time may only be increased by increasing the thickness (and consequently the bulk and weight) of the suit. On this basis it appeared that the only practical aprpoach was to select the best possible material, to design a suit to make the most effective use of its properties, and, finally, to conduct practical trials of the completed garment to determine the exposure time permissible in the garment.

The additional requirements, that the suit be impermeable to steam and that it be relatively comfortable, need little discussion, except to point out that they are also in some measure contradictory. A suit which will not permit steam to

enter will necessarily tend to be uncomfortable, since it will prevent the evaporation of perspiration ,and the discomfort will be increased, especially in warm weather, by the insulation needed to protect against high temperatures.

Based upon the study of all of these factors, tentative requirements for an acceptable material were proposed. These were as follows:

1. The fabric, or combination of fabrics, must be capable of withstanding a temperature of 2000° F., on its exterior surface at least.
2. It must be high in insulating value.
3. It must be flexible and comparatively light in weight.
4. It must be impermeable to vapors.
5. It must be capable of fabrication into a garment by sewing or other practicable means.

SELECTION OF MATERIALS

With these requirements established it was immediately apparent that no single fabric was likely to combine all of the desired properties. The greatest hope of success appeared to lie in finding or developing a combination, each member of which should satisfy one or more of the requirements. It was postulated that this combination should consist of at least four layers, as follows: (1) an outer cover of strong, abrasion- and heat-resistant fabric; (2) a layer, or layers, of material having high thermal insulating value; (3) a vapor barrier, such as a coated fabric; and (4) a lining providing additional mechanical strength and contributing to the comfort of the wearer.

During the course of the investigation a great many materials have been considered for possible application in the suit. Most of these were discarded without actual test. In this category were flameproofed organic fabrics, such as fire-resistant cotton duck, proposed for use as a cover material. Such materials were eliminated in the belief that the exposure conditions prescribed were so severe as to be met only by completely incombustible inorganic fibers such as glass and asbestos.

In the final selection, a simple screening test, developed by the Laboratory, proved useful. In brief, this consisted in mounting a 6- by 8½-inch specimen of the composite fabric in the one open side of a small rectangular sheet-steel chamber, 6 by 8½ by 4 inches in size. The chamber was insulated with ½-inch thick fibrous glass to prevent heat loss, and was provided with openings for the introduction of thermometers by which the temperature of the air within the chamber and on the inner surface of the fabric could be measured. The exterior surface of the fabric was exposed, through a 2-inch spacing sleeve, either to the open door of a muffle furnace maintained at any desired temperature (usually 1800° F.), or to an auxiliary chamber filled with saturated steam. Provision was made for a spray tube to be inserted through holes in the spacing sleeve so that water could be directed along the entire upper portion of the test fabric. Heat transfer through the fabric combination was determined by frequent readings of the two thermometers; comparative evaluation of two fabrics was based on the relative time required for the interior temperature to reach 130° F. in each case. This relatively crude test proved useful, not only in the selection of materials, but also in determining the best arrangement for those in the combination.

As indicated, fabrics composed of flame-proofed organic fibers were eliminated from consideration as cover materials without test, since, although they will not burn, they decompose and disintegrate at temperatures far below the prescribed 2000° F. Strictly speaking, on this basis glass and asbestos fabrics

should also have been eliminated, since they lose strength and become brittle when exposed for any length of time to temperatures above 1000° F. However, discarding these would have left only metallic fabrics, most of which are relatively stiff, high in heat conductivity, and poor in flex resistance. For this reason glass and asbestos were retained, with the expectation that in actual service a single exposure to the most severe conditions might render the suit unfit for further use. Both glass and asbestos fabrics have been tried in actual suits. Asbestos is better in durability, but is heavier, absorbs more water when wetted, and presents a rougher appearance.

For thermal insulation, felted batts of glass, asbestos or rock wool appeared most promising. Reflective barriers such as aluminum foil were also considered. Among the felts, superfine fibrous glass acoustical insulation was found most effective. This is readily available commercially and has several advantages in addition to its excellent insulating value. It is provided with a silicone-resin-water-repellent treatment, which prevents absorption of large quantities of water in case the suit is wetted down in service, and the fiber is so fine as to cause no skin irritation, which is a problem with all other fibers considered. Two layers of this material, in 1-inch nominal thickness, combined with a layer of aluminum foil, were found in the laboratory to give better performance than any other combination of equivalent thickness. Subsequent trials in an actual garment have, however, demonstrated that the foil is impracticable for use, since it disintegrates rapidly in service. For this reason, two layers of fibrous glass only were the final selection.

As a vapor barrier a lightweight glass cloth coated with neoprene was selected. This material leaves much to be desired, since the neoprene coating, although non-inflammable, is decomposed by heat. This is important, since in the laboratory screening tests exposure to steam demonstrated that the most effective location for the barrier was immediately beneath the outer cover, as far as possible from the body. However, no better material has been found, and therefore, as a further compromise, the barrier has been given some protection by locating it between the two layers of fibrous glass insulation.

A white woolen flannel was selected for use as a lining. This material is also combustible, although not highly inflammable. However, it is not exposed to direct heat at any point, and, indeed, if it became so hot as to ignite, the wearer would be in danger whatever the type of lining. It provides some additional insulation and absorbs perspiration, contributing in this way to the comfort of the wearer.

The combination of fabrics finally selected was, then, as follows:

1. An outer cover, consisting of either glass or asbestos cloth.
2. A layer of 1-inch fibrous glass insulation.
3. A vapor barrier of lightweight neoprene-coated glass fabric.
4. A second layer of 1-inch fibrous glass insulation.
5. An inner lining of woolen flannel.

DESIGN AND CONSTRUCTION OF THE FIRE FIGHTERS' SUITS

After the fabric combination had been decided upon, the limited available literature was studied, and experienced fire fighters were consulted to determine the defects of the current asbestos suits and improvements desirable in any new design. In this phase, as in the subsequent service trials, the advice and active cooperation provided by personnel of the Naval Damage Control Training Center, and in particular those of the Fire Fighters' School, proved invaluable. In the early discussion it soon became apparent that the one-piece con-

struction of the standard garment was not suited to the new materials, nor to the service requirements for the new suit. The vapor barrier included in the new material made the provision of an enclosed air supply obligatory; if this were to be the Oxygen Breathing Apparatus, as seemed most logical since this device is a standard item of Naval fire fighting equipment, this must be wholly enclosed within the suit to protect its vulnerable rubber parts from the intense heat. This would be difficult, if not impossible, in a one-piece garment. Zippered openings, as used in the former suit, could not be tolerated in a garment designed to be impermeable to vapors, and the zippers themselves would be difficult to protect from flames. On the other hand, attached gloves and shoes, a feature of the asbestos suit, were considered desirable in any new garment.

Following these discussions, a preliminary design for the suit was drawn by the Laboratory, and a suit was constructed at the Shipyard Sail Loft. This first model was constructed of the combination of fabrics previously described, with an additional layer of aluminum foil placed between the outer layer of insulation and the vapor barrier. It was critically examined and worn by personnel of the Laboratory and of the Fire School to evaluate its weight, comfort and general utility. Finally, the suit was worn briefly in an actual fire by the Officer-in-Charge of the Fire School. In this manner a number of deficiencies were found, among them the tendency of the aluminum foil to disintegrate rapidly in use. The foil was omitted from subsequent suits, and a number of minor changes were made in the design, principally to allow greater freedom of motion to the wearer. In most of its essential features, however, the present design is similar to that first prepared.

The present suit is composed of three separate units. The first of these is a bib-type overall, with attached shoes, over which the standard Oxygen Breathing Apparatus is worn. The bib protects the wearer's chest from the heat liberated by the canister of the O. B. A., and provides double protection over the front of the body. Take-up straps of cotton webbing on each side permit adjustment to the wearer's waist, and asbestos tape draw-strings are placed at the ankles to hold the shoes firmly in position. Draw-strings are necessary at this point, since the suit is designed to be worn over the fire fighter's normal clothing, including his regular shoes, and must be loose enough to be donned quickly by a wearer of any ordinary size. The shoes are constructed of the same material as the rest of the garment, with an added outer sole of non-skid turret deck covering fabric. The latter is a cotton fabric, with a heavy flameproofing treatment, and provides a durable, non-slip wearing surface. An adjustable harness of heavy cotton webbing, extending through the crotch, is built into the interior of the trousers, and terminates in a steel "dee" ring located between the shoulder blades. A wire braid, insulated by an asbestos cover, extends from this ring through slits in the jacket and hood to the outside where a second ring is attached. This may be connected to a steel cable which serves as a life line. Figure 1 shows a fire fighter dressed in the overalls, with the O. B. A. in place. This photograph also shows the wire guards which have been added to the O. B. A. to prevent collapse of the rubber air sacks when worn under the jacket of the suit.

The second unit of the suit is a parka-style jacket with gloves attached, shown in Figure 2. The parka is designed to fit snugly about the face-piece of the O. B. A. The jacket must be wide at the bottom in order to pass over the O. B. A., and a draw-string is placed at the lower hem to close the wide opening at the waist once it is in place. Elastic wrist bands in the sleeves hold the gloves in position on the hands.

Figure 1.—Photograph of a fire fighter clothed in the trousers, with the Oxygen Breathing Apparatus in position. Note the attached shoes and the protective wire framework which has been added to the canister holder of the O.B.A. to prevent collapse of the rubber air sacks when worn under the jacket.

Figure 2.—This picture shows the fire fighter with the parka-style jacket in place. Note that the Oxygen Breathing Apparatus is completely enclosed and protected, except for the rubber face piece which is subsequently covered by the hood.

The gloves are constructed in the form of mitts; separate fingers are almost impossible to fabricate from the bulky fabric combination without undue compression and consequent loss of insulating value. The palms are faced with chrome-tanned cowhide.

The final unit of the suit is the hood. Figures 3 and 4, photographs of a fire fighter completely dressed in the suit, show this in place. It has a skirt which extends down over the shoulders. When worn over the parka, the helmet thus doubles the protection over the most vulnerable portion of the body. A special heat-resistant glass known as "Vycor" provides vision. In the suit shown in the photograph a single panel of glass was used; the latest suit has two panels mounted side by side to give a wider field of view. It is proposed, in the future, to include a fiber skull guard of the ordinary "hard hat" type in the hood to provide additional head protec-

Figure 3.—Photograph of the fire fighter completely dressed in the suit. In later models, vision has been increased by mounting two sections of heat resistant glass vertically in the hood to replace the single, horizontal plate.

tion. Figure 4 also shows the insulated strap and ring to which the life line is attached. Since this photograph was taken the strap has been shortened so that only the ring appears on the outside of the hood.

In the fabrication of all units of the suit the felted fibrous glass insulation is held in place by loosely quilting the two layers and the vapor barrier material to the flannel lining. All seams are made as impermeable as possible by turning and covering with asbestos tape. The total weight of the entire outfit, including the O. B. A. is slightly over 40 pounds, varying to some extent depending upon the cover fabric used. Approximately 14 pounds of this total is contributed by the O. B. A.

No accurate cost data are available, since no suits have been manufactured commercially. However, judging from experience gained at the Shipyard Sail Loft, it is estimated that the complete outfit can be produced in reasonable quantities at a cost of approximately $200.00.

Figure 4.—Rear view of the fire fighter, completely dressed. In later models, the insulated strap which extends down the back has been shortened so that only the ring protrudes. The strap is connected on the inside to the built-in rescue harness, and the ring may be attached to a life line.

SERVICE TRIALS

Since the first prototype was constructed, three additional suits have been manufactured. The first suit was given only a limited trial; the three additional suits have been tested under the most extreme conditions which it has been possible to devise. Two have been worn until, after repeated exposures, they were no longer serviceable. In this phase also, the advice and active cooperation of the Fire School personnel have been invaluable.

The preliminary tests were quite simple, and only involved donning the suits, with the Oxygen Breathing Apparatus, and wearing them for increasing periods of time with increasing physical activity. Both Laboratory and Fire School personnel participated in these tests. No attempts were made at first to do more than wear the suits and become familiar with their characteristics; later, activity was increased until the wearers were able to climb ladders, to run, and to manipulate tools.

Temperature measurements were made on the air which the wearer drew from the O. B. A., since the canister of this apparatus generates a great deal of heat which, it was feared, would not be readily dissipated when enclosed within the suit. The harness for attaching the life line was also tested at this stage by substituting tackle for the life line and lifting the wearer from the ground by this means. At this time also, limited tests were conducted to determine the length of time during which the suit might be worn without the O. B. A., with the wearer breathing only the air within the suit.

The first trials served to familiarize personnel with the characteristics of the suits, and to overcome any claustro-

phobia which might be felt in donning them. They also demonstrated that the suits place some strain upon the interpretation of the word "relatively" in the requirement that they be relatively comfortable. As was expected, in hot weather they are not comfortable in any true sense of the word, and wearers perspire freely. However, the flannel lining absorbs the perspiration, and the suits are probably as little uncomfortable (to describe them more accurately) as they can be made in consideration of the protection which they must provide. Suits have been worn continuously for periods of 30 minutes in very hot weather, and this wearing has been repeated a number of times by the same subject on the same day without ill effects. The 30-minute time limit for a single wearing is set by the O. B. A., rather than by the suits, since this is the safe useful life of a canister of this device when the wearer is exercising.

On the positive side, these trials demonstrated that, although somewhat cumbersome, the suits are wearable and that, in a suit, the wearer can perform

Figure 6.—Fire fighters running while wearing the new suits.

Figure 5.—Fully clothed fire fighters demonstrating the maneuverability of the suits by climbing ladders.

any tasks which are likely to be required of him. He can walk, run, climb ladders, handle hose lines, open and close valves, and use such tools as wrenches, axes, crowbars and the like. Figures 5 and 6 show fire fighters climbing ladders and running, respectively, while dressed in the suits. The Oxygen Breathing Apparatus functioned satisfactorily within the jacket, and the temperature of the inspired air was not significantly increased when the canister was so enclosed. In an extreme emergency, a fire fighter can even use the suit for as much as 3 or 4 minutes without the O. B. A., breathing only the air trapped within the hood. The rescue harness also performed as intended; Figure 7 shows a fire fighter freely suspended by it.

After the subjects had become thoroughly accustomed to the suits, actual service exposures were initiated. In all of these tests the suits were worn only by experienced fire fighters from the School. As in the previous tests, these exposures were of graduated severity,

starting with those which were relatively mild and culminating in the wearing of the suits in fires of high intensity.

First among the trials was one in which the suits were worn in a steam-filled atmosphere. For this test all doors and bulkheads of the simulated engine-room compartment of one of the training structures at the Fire School were closed, and saturated steam was admitted through three two-inch lines. The room was preheated for about 20 minutes and the temperature rose to 185° F. in this time. Two fire fighters, clothed in the suits, then entered the compartment and remained inside for 10 minutes, walking about and climbing up and down ladders. Neither subject, during this period, experienced more than a slight sensation of warmth, and this was confined to small areas of the body, as along the arms.

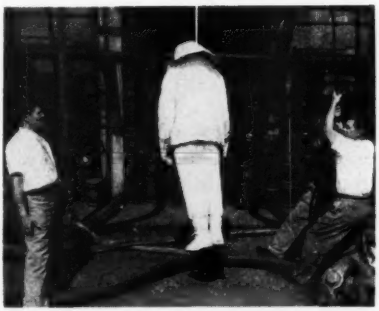


Figure 7.—A fire fighter suspended by means of the rescue harness and life line attachment.

The steam exposure was unspectacular, but served to build confidence in the wearers, and demonstrated that at least one of the deficiencies of the previous asbestos suit had been overcome.

Figure 8.—Two fire fighters wearing suits enter a gasoline-spill fire. The shortened life line attachment can be seen in this photograph.

Figure 9.—A fire fighter emerging from a gasoline-spill fire.

Low as the actual temperatures were, no experienced fire fighter would have considered entering the steam-filled atmosphere while wearing the former garment.

The suits were next tried in flame. Fifteen gallons of gasoline were poured out on an open paved area and ignited. Two fire fighters were involved in the test, one wearing a suit which had been wetted down, the other wearing a dry suit. The first wearer entered the fire and walked back and forth through its center for approximately two minutes; the second remained near the periphery of the area and operated a hose equipped with a fog-spray nozzle, with which he eventually extinguished the flames. Neither fire fighter experienced discomfort in the suits, and the latter were undamaged, although the heat was intense and flaming gasoline was picked up by the shoes and splashed upon the legs of the wearers. Wetting down the first suit moderated the exposure somewhat, but provided additional evidence of the effectiveness of the vapor barrier, since the water was completely vaporized by the heat during the trial.

Such fires have been repeated many times and with a number of variations. Figures 8, 9, and 10 are photographs of fire fighters entering and emerging from flames in three of them. Quantities of gasoline have been increased to as much as 100 gallons; dummies clothed in asbestos have been placed in the center for rescue by the fire fighters; and suits have been worn both wet and dry. In a recent demonstration, which was held by the Damage Control Training Center for interested Naval, Army, and civilian personnel and for the press, a scrapped airplane was placed in the center of a 100-gallon gasoline fire, and

fire fighters climbed upon the wing to rescue a dummy pilot from the blazing cockpit. The fires are always gratifyingly spectacular, and the heat is so evident to observers that the effectiveness of the suit is immediately apparent. Visual evidence of the intensity of the heat is provided by Figure 10, in which a fire fighter in ordinary clothing, manning the hose line, can be seen shielding his face. However, increasing the quantity of gasoline has not materially increased the severity of the exposure as far as the suits and their wearers are concerned; gasoline burns at approximately the same temperature regardless of the quantity involved. The first test, therefore, demonstrated the effectiveness of the suit as well as those which followed, even though greater quantities of gasoline have since been used. The principal value of the subsequent demonstrations has been to provide information as to the durability of the suits in repeated wearings.

Following the test in the gasoline-spill fire, the two fire fighters again wore the suits into the engine-room compartment which was used in the steam test. In this case, however, the compartment was flooded with a blazing mixture of gasoline and fuel oil, and the fire fighters walked over gratings above the flames. They remained in the space for approximately one minute. Although this test was less spectacular, the exposure is believed to be even more severe than that in the open, since the flames were confined and the fire fighters were required to walk above them. Temperatures, measured at several points within the space, were found to approach 2000° F. This demonstration also has been repeated a number of times, with dummy rescues in several instances.

DISCUSSION

The most important characteristics of the new suits, which has been demonstrated in all of the service trials, is the high degree of protection which they provide. In only one of the many tests has the wearer reported more than a slight sensation of warmth. The single exception occurred in one of the gasoline-spill fires, in which one fire fighter received a first-degree burn on a small area of his right leg while carrying a dummy from the flames. This may have resulted from his clutching the dummy, which had become saturated with gasoline, tightly against his body, compressing the fabric with reduction of its insulating value. It has frequently been observed that there is a sensation of heat within the suit when the material is compressed, as when leaning against a hot object in a fire, or when sharply flexing a knee or an elbow. It is also possible that the insulation of this particular suit, which had been worn in at least 15 previous trials, had become thinned at points of flexure. Whatever the cause, the burn was minor, and the wearer, after changing his suit, was able to proceed with further trials on the same day.

The maximum duration of any flame exposure has been approximately two minutes. In every test the exposure has been terminated before the wearer became uncomfortably warm, except for the case just mentioned, although the fire fighters have frequently stated their willingness to remain longer in the flames. It is thus impossible to fix definitely a safe time limit for wearing the suit. The limit is probably about three minutes for gasoline fires. Lower exposure temperatures will, of course, extend this time. Three minutes is a relatively short period, but should be sufficient to rescue a victim trapped in a burning plane, or to perform many emergency tasks. If the job cannot be done in this time, it must await extinguishment of the fire, or must be done in relays.

Figure 10.—In this photograph two fire fighters are simulating the rescue of a pilot from a crashed airplane. In actual practice, of course, a blanket of material similar to that of the suit would be needed to shield the victim from the fire. The fire was produced by 100 gallons of gasoline. Note that the plane is completely obscured by the flames; the heat can be judged by the manner in which the second hoseman is shielding his face.

One gratifying result of the service trials has been to establish the fact that the suits have a much greater service life in fires than was anticipated. As was mentioned previously in the section devoted to the selection of materials, glass and asbestos fibers both lose strength and become brittle when exposed for any length of time at temperatures above 1000° F. For this reason they would not even have been considered had suitable materials of greater heat resistance been available, and, because of their deficiencies in this respect, it was expected that garments made from them might be rendered unfit for further use by a single exposure. However, actual suits have been worn in fires repeatedly, some for as many as 15 times, without serious failure of the outer fabric. Embrittlement and loss of strength have occurred only in small areas where gasoline has burned directly on the surface, or where heat has been concentrated by some other means. Even at such points the fabrics have retained sufficient strength to hold the insulation in place, so the security of the wearer is not endangered. The chrome leather facings for the gloves, and the treated cotton shoe soles have also proven to be surprisingly durable. The only apparent explanation for this fortunate development is that there is a greater time lag than was anticipated, and that the fabric does not approach

the flame temperature in the actual time of its exposure.

The time required to don the complete suit, including the Oxygen Breathing Apparatus, is between three and five minutes. This is somewhat high for a garment designed for use in emergencies. However, since a large part of this time, probably at least half, is required to adjust and activate the O. B. A., it is not likely that it can be reduced greatly by refinements in design of the suit. In regularly scheduled operations, such as landings and take-offs on board an aircraft carrier, the practical expedient would be to have fire fighters operating in relays, with at least one standing by fully dressed at all times. If this is not practicable, having the fire fighter partially dressed, wearing trousers and O. B. A., but with the face piece of the latter dropped, might serve the purpose. However, there appears to be no immediate answer for the emergency which cannot be anticipated.

The best recommendation which can be made for the new suit and the adequacy with which it fulfills a real need, is the enthusiasm with which it has been adopted by experienced fire fighters. Personnel of the Damage Control Training Center, and in particular of the Fire Fighters' School, have been its most ardent champions from its inception. The latter have not only spoken in its favor; they have repeatedly risked their own lives in demonstrating its performance. Indeed, their enthusiasm has been a source of concern to those of the Laboratory staff who have stood by during the many fire tests. It has often been difficult to restrain the fire fighters from prolonging the exposures unduly and to keep them from taking unnecessary risks. This enthusiasm has been shared to a considerable degree by civilian fire fighters since the public demonstrations previously mentioned, and numerous inquiries have been received as to when the suits will be available commercially. The demonstration, which was reported in the daily press, newsreels, and "Life" magazine, was in itself an indication of the fire-fighters' attitude, since it was wholly conceived and executed by the Damage Control Training Center in an effort to publicize the suit and speed its acceptance. With this attitude on the part of practical fire fighters, the Industrial Test Laboratory feels that, although the suit may be improved by further development work now underway, the major objective has been achieved, and that the Navy has available to it a suit which is practical, usable, and which far surpasses in performance any previous equipment of this type.

ACKNOWLEDGMENTS

The writers wish to acknowledge their very great indebtedness to Captain Charles E. Harrison, U.S.N., Commanding Officer of the Naval Damage Control Training Center at the Philadelphia Naval Base, who not only made available the facilities of the Fire Fighters' School for testing the suits, but also cooperated actively in their design. He conceived and directed the demonstration on 15 September 1949 at which the suits were shown to officials of the Navy and of the other Armed Services as well as to the public. Lieutenant Commander A. B. Mullen, U.S.N.R., Officer-in-Charge of the Fire Fighters' School, and Carpenter W. B. Haselton, Jr., U.S.N., were not only of great assistance in the solution of design problems, but also wore the suits in all of the service trials. The debt to these two needs no further emphasis than to state that they risked their own lives in these demonstrations.

Captain E. R. Sperry, U.S.N., and Lieutenant Commander E. W. Phillips,

U.S.N.R., of the Damage Control Section, Bureau of Ships, were active in their support of the project. Mr. R. W. Webster, of the Research and Development Branch of the same Bureau, supplied many of the original concepts and contributed much to the development at all stages.

Lieutenant J. K. Coates, U.S.N., formerly Superintendent of the Industrial Test Laboratory, his successor in this position, Commander W. E. Pittman, U.S.N.R., and Mr. J. E. McCambridge, Superintending Engineer, contributed valuable advice and support. Mr. H. G. Kime, former Head of the Textile Section, directed the initial planning of the project, which was completed under the direction of Mr. S. Schneider. Mr. D. Nusbaum, of the Protective Clothing Group, contributed to the design, conducted the laboratory tests, and prepared drawings. Messrs. A. J. Dowling, E. Urbanski and A. Grill, of the Shipyard Sail Loft, also contributed valuable advice on the design and developed practical methods of fabrication.

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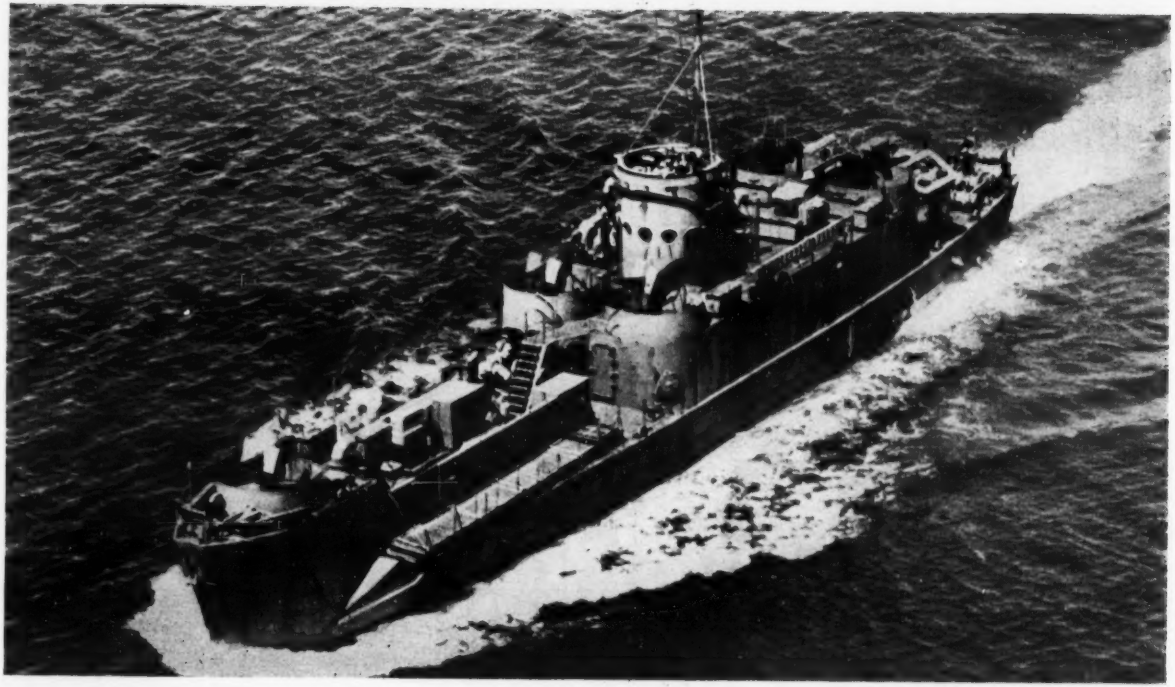


Official U. S. Navy Photograph

Oblique of the LCI 435.

AN UNUSUAL METHOD OF MAIN ENGINE ALIGNMENT

LAURIE R. WALLACE

THE AUTHOR

Early training consisted of seven years in the design section of Morse Dry Dock and Repair Co., Brooklyn, N. Y., special courses in Naval Architecture and Marine Engineering at Pratt Institute, Brooklyn, N. Y., about 16 years in Building Construction. From early 1937 to date, Civil Service with the Navy, seven years of which was in the Design Section Engineering at the Brooklyn Naval shipyard and since February 1943 to date in the office of the Assistant Industrial Manager U. S. N. 6 N. D., Jacksonville, Florida. He is a member of the Florida Engineering Society, The Propeller Club, Port of Jacksonville and an Associate Member of the U. S. Naval Institute. His home is in Jacksonville, Florida.

During the early part of May 1943 the U.S.S. *Absecon* (AVP-23), operating off the mouth of the St. Johns River, Florida, grounded momentarily on a submerged hulk. Upon a dry dock inspection immediately following the accident it was found that the shell plating between the keel and the starboard bilge in the vicinity of the engine room had been crushed up into the fuel and water tanks, some of the longitudinals and transverse members had been bent, torn, and distorted to the point of deflecting the tank top under the starboard inboard engine foundation. The corresponding deflection of the engine foundation had moved the engine up and outboard.

The main propulsion plant of this vessel consists of four ten-cylinder opposed piston Fairbanks-Morse diesel engines, each pair coupled to one shaft through oil clutches and reduction gears. As mentioned above it was the inboard engine of the starboard pair that was directly above the damage to the outer shell plating. From readings of a crankshaft strain gauge between the webs of the crankshaft at number ten crankpin before and after coupling the engine, it was apparent that a serious misalignment of this engine existed. Upon checking the engine to reduction gear coupling after uncoupling it was found that the whole engine had been moved outboard .030 inch, and except for a slight upward movement, was otherwise within tolerance limits.

To put this engine again in alignment with the reduction gear and shafting meant either removing all 48 body bound bolts, reaming for oversize bolts and spotwelding all the new steel chocks in place after the long, expensive operation of hand fitting, milling, filing, blueing and dressing to size with feeler gauges or rigging jacks at about six places along both sides of the engine foundation and jacking the whole foundation together with the engine back into place and true alignment. The writer proposed the jacking method for three reasons as follows: First, it would save precious time; second, it would effect a large saving in cost; third, previous experience of a similar nature.

At a conference consisting of the Assistant Industrial Manager, 6th Naval District, Jacksonville, Florida, the Commanding Officer and the Engineer Officer of the vessel, representatives from the Merrill-Stevens Dry Dock & Repair Co., Jacksonville, Florida, the engine manufacturer's local engineer, and myself, called to determine and act upon the best method of realigning this engine it was brought out that the jacking method of realignment of an engine had never been heard of before. However, in view of the fact that it would take not less than fifteen days and nights to disconnect and raise the engine, make new chocks and body-bound bolts, realign engine and connect up in running order against one day and night for the proposed jacking method it was decided to first try the jacking even though a majority of those present were of the opinion that permanent alignment could not be maintained by this method.

So upon orders to proceed with the jacking method a start was made at 1600 Saturday, May 22, 1943.

The sketch shows the location of the jacks and blocking as disposed between the foundation members and also the new angle iron stiffener welded to the inboard brackets after alignment was accomplished.

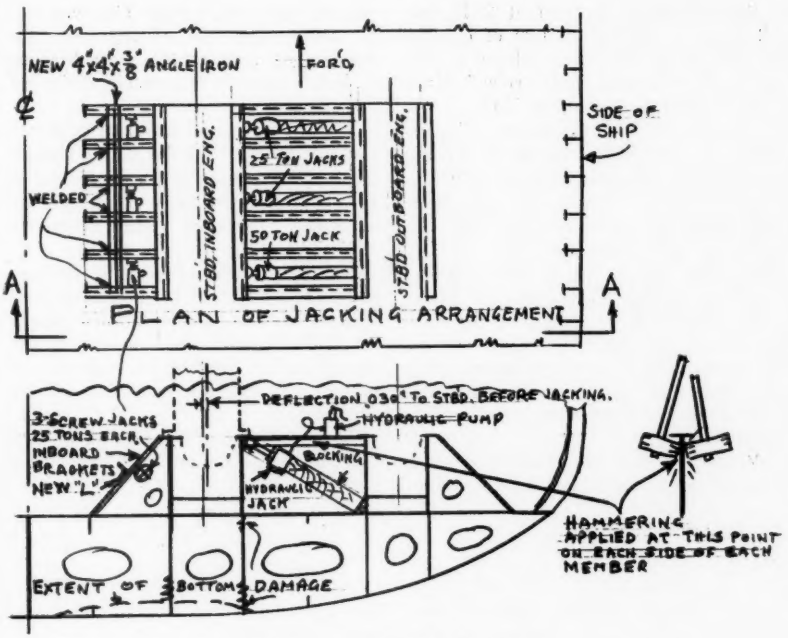
By 2200 Saturday night everything was in readiness for the operation. On Sunday morning, 23rd of May, dial indicators were placed on the misaligned couplings and the strain gauge was again set up between number ten crankpin webs on the crankshaft.

By 1000 all was in readiness for a test strain on the jacks. The first strain on the jacks resulted in moving the engine and foundation to within .008″ of true alignment on a dial gauge reading of .022″ +. Note that the original misalignment was .030″, so it must be assumed that placing the jacks and taking them up hand tight in place had even with this small strain, moved the foundation from the original distortion of .030″ to .022″, some of this movement may be attributed to a certain amount of natural stress relieving due to the act of cutting out damaged shell plating, repairing the damaged longitudinals and transverse members of the double bottom and testing the compartments below the foundation of the engine. When this first strain on the jacks was released the engine and foundation moved back to .010″ thus making a net gain of .012″.

(At this point arrangements were made to hold the engine in alignment if the jacks were able to move it inboard the full amount to true alignment while a thorough check was made of the foundation weldments to detect any possible fractures at the joints with the tank top.)

This was done by clamping the new angle iron stiffener fore and aft along the outer edges of the inboard brackets of the foundation as shown on the accompanying sketch.

The jacks were again worked to move the engine foundation inboard to a point .004″ beyond the required amount. To relieve the tension strains in the transverse members between the inboard and outboard engine foundations caused by the pressure of the jacks, simultaneous pounding by large hammers was resorted



Section A-A Looking Forward
(not to scale)

to on all six members hitting both sides of each steel member about ten solid blows each with two hammers pounding the same member at the same instant. This was equivalent to elongating the members and at the same time relieving internal tension strains. While the hammering was in progress the jacks were slacked off while a constant check was made on the dial indicators. During this time the inboard jacks were tightened up to hold the gain made by the outboard jacks and the hammering. The angle iron was then welded to each bracket to hold the gain. When the welds were cooled the three 25-ton screw jacks were then released from between the inboard brackets, leaving the engine within allowable tolerance of alignment. Final readings on the coupling flanges showed concentric within .006" and parallel within .003" stub shaft thrust collar clearance .006".

Monday, May 24, 1943, checked alignment of gear teeth in reduction gear case with blue and found same in perfect alignment. Engine was then coupled up and strain gauges were checked on all crankshaft throws. All that showed strains were well within manufacturer's tolerance limits.

May 25, 1943. Warmed up all engines at 0831 to one-third speed. Left the shipyard at 0831, arrived fuel oil dock at 0900, all engines working well, bearings cool at 95 r.p.m. on shafts and up to 130 r.p.m. in reverse, secured at 0920.

Started taking on fuel at 0935, left oil dock at 1345. Both sets of engines at 2/3 speed, 500 r.p.m., shafts at 195 r.p.m. all bearings cool, arrived Mayport, Florida Section Base 1545.

May 26, 1943. Left Mayport May 26 0400, all engines full speed, shafts 240 r.p.m., 18 knots, cylinder temp. 600° F., engine bearings normal, seas mild. Eng. and Reduction gear bearings 105° F. Lube oil from engine 110° F. Off Charleston, S. C. Light 1315, arrived Charleston Navy Yard 1530.

After engines were secured, the starboard inboard engine was again opened up and crankshaft strain gauge readings were taken, all readings showed strains well within the manufacturer's allowable tolerances.

You will notice that frequent mention is made of the main shaft bearings during this post repair trial. This was done to assure that no misalignment had occurred to any of the shaft bearings or bulkhead seals as a result of the bottom damage.

The engine manufacturer's representative was on board all during the realignment of the engine and accompanied us to Charleston. He was well satisfied and agreeably surprised that everything worked out so well.

The ship's officers and the management and workers of the Merrill-Stevens Dry Dock Co. all cooperated wholeheartedly towards the success of this operation. Six months later all was reported well on the operation of this engine.

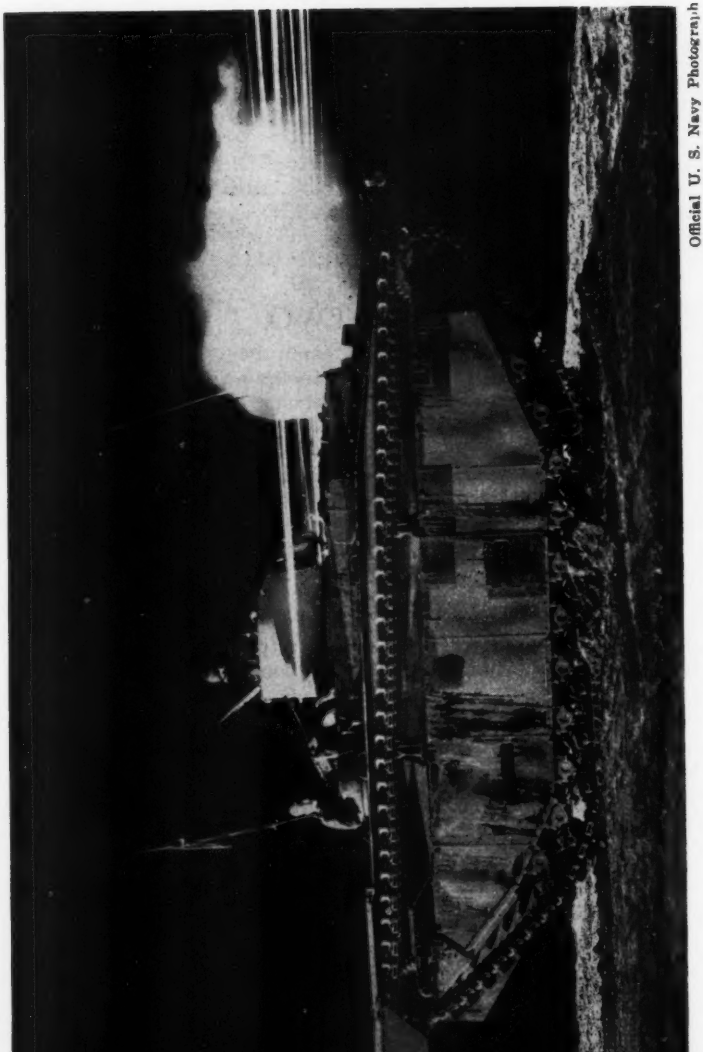
To make a long story short we bent the engine foundation back to where it belonged.

PART TWO

REPRINTED ARTICLES

The articles contained in this Part Two represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the JOURNAL, for articles originally published elsewhere which contribute to the purpose of the Society: to further the advancement of naval engineering.



Official U. S. Navy Photograph

"Water Buffalo" Joined the Fleet during World War II.

MARINE BOILER DETERIORATION

ACKNOWLEDGMENT

This article, by DR. I. G. SLATER, Director of Operational Research, Admiralty, and MR. N. L. PARR, Senior Scientific Officer, R.N.S.S., surveys the several forms of boiler corrosion and deterioration encountered in Marine practice. Several forms of the various phenomena are illustrated and basic principles discussed. The article appeared in the Journal of the Institution of Mechanical Engineers for December 1949.

INTRODUCTION

During the 1939-45 war, a full opportunity was made available to study instances of boiler corrosion and deterioration which occurred in His Majesty's ships while they operated under adverse conditions. In marine boilers, adverse factors seldom encountered in normal land practice are commonplace. The sea permeates relentlessly every nook and cranny inboard. To maintain the purity of boiler feed water is a difficult task, and it is to the credit of the marine engineer that the troubles of boiler operation and maintenance have been so far overcome. In addition, circumstances have dictated the rapid adoption of many new modifications and designs in boilers, most often with the view to increasing performance per unit weight, to improve flexibility and to economize in ship's space.

Forms of corrosion and deterioration which may be encountered in marine boilers of both water-tube and fire-tube types have been classified in Table 1; a key to photographic illustrations of each form of corrosion or deterioration, and also a description of their respective characteristics and causes are given.

The sections which follow discuss in appropriate detail each of the types of corrosion and deterioration noted, and outline certain experimental evidence which has been accumulated in an attempt to elucidate the more fundamental factors involved. An Admiralty pattern boiler is shown in Fig. 1. Published work in the field of corrosion has been freely drawn upon in describing the various phenomena involved, and the authors are particularly indebted to the monumental contributions of Dr. U. R. Evans and his associates at Cambridge University.

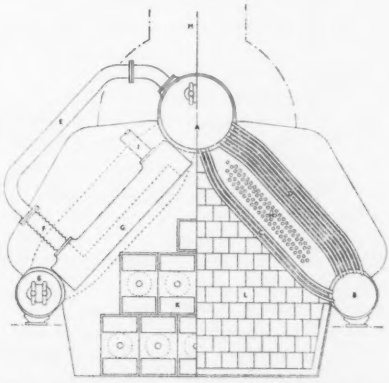


Fig. 1. Marine Water-Tube Boiler, Admiralty Yarrow Type

A Steam drum.
B Water drums.
C Generator tubes.
D Downcomers.
E Saturated steam pipe.
F Corrugated expansion pipe.
G Superheater header.
H Superheater tubes.
J Main steam line.
K Oil burners.
L Furnace.
M Uptake.

"Water Buffalo" Joined the Fleet during World War II.

TABLE 1. FORMS OF BOILER DETERIORATION

Classification			Appearance of corrosion	Usual location	Cause	Fig. No.
Boiler Deterioration — Internal Deterioration — Pitting		Air bubble pitting	Isolated hemispherical pits, often covered with a cap of loose soft corrosion products, which when dry is of black iron oxide covered with a membrane of red dehydrated ferric hydroxide.	Mainly on the roof of steam and water drums, but likely to be found on any surfaces in contact with water, where air bubbles may collect.	Liberated air bubbles attaching themselves to metal surfaces, thereby initiating anodic attack by the generation of localized electrolytic action.	3 4
		Scab pitting	Deep localized pitting covered by a scab of hard scaly black magnetic iron oxide which cannot be removed by wire brushes.	In hotter areas of heating surfaces where corrosion products are formed and rapidly consolidated into a hard scab by the heat. This is possibly the main form of deterioration encountered.	Use of untreated impure boiler water containing dissolved salts and gases in comparatively small concentrations so that pH of water lies between 4·8 and 9·4.	6 50 7 51
		Soft scab pitting	Broad, shallow pitting covered with caps of fairly hard corrosion products, usually black magnetic iron oxide covered with a layer of red dehydrated ferric hydroxide.	In superheater tubes and headers and in main steam lines.	Deposits of salts by priming in superheaters, main steam lines, etc., followed by condensation when the boiler is shut down, resulting in the formation of droplets of electrolyte and the initiation of a pit by differential aeration.	9 10
	General Wastage	Tube wall thinning	A general reduction in tube wall thickness over comparatively large areas, often without the presence of blankets of corrosion products.	Fairly evenly distributed over water surfaces with a tendency to be more pronounced in hotter areas.	Use of acid feed water.	12 13 14
		Necking and grooving	A circumferential band of corrosion on the water-side of fire tubes, stay tubes, and stay bolts in Scotch boilers.	Where stay bolts, stay tubes, and fire tubes enter end plates or combustion chambers.	The influence of adjacent end plates on expansion and contraction, resulting in stresses which cause repeated localized flaking of protective films.	15 16
		Galvanic attack	Local reduction in metal thickness.	Adjacent to non-ferrous boiler mountings, in areas of non-homogeneity of steel composition or at breaks in surface films.	Use of dissimilar metals in contact (or at areas of non-uniformity on the metal surfaces) in the presence of an electrolyte, when a difference in potential will result in sacrificial electrolytic attack at the anode.	17 53
	Corrosion Fatigue	In tubes	Network or series of cracks over the water-side of tube surfaces.	In boiler and superheater tubes on the hottest side of the bores. It is most frequently found in inherently curved tubes or tubes which have been buckled in service.	Irregular circulation in tubes causing frequencies of alternate over-heating and cooling which induce pulsating thermal fatigue stresses in the walls. Cracking is accelerated by residual stresses and a corrosive environment.	19 20 21
		In steam and water drums	Fissures, usually difficult to detect due to covering layers of corrosion products; usually associated with deep pitting.	Commencing at the arbouring in tube plates and extending axially, eventually joining two or more holes.	Bending stresses due to lack of common neutral axis between tube and wrapper plate, and excessive cold working due to initial rough machining operation. Accelerated by thermal stress and corrosive environment.	22 23
		In main steam pipes	Lines of cracks on bores of pipes.	On the bores of main steam pipes, especially in the base of corrugations.	Bending stresses due to movement during operation in the presence of a corrosive environment.	24
		Strain age embrittlement	Spontaneous cracking in boiler components during detubing or cleaning operations.	Between tube holes in tube plate, under rivet heads, round top end of tubes adjacent to the rolling surfaces.	Reduction of impact properties by the combination of cold work and ageing. Spontaneous failure occurs only at and below room temperature.	25
		Caustic embrittlement	Fissures in boiler plates, etc.	In boiler plates, radiating from and often connecting adjacent rivet holes, joints, arbourings, etc.	Joint effect of a combination of non-uniform stressing, and high concentration of caustic soda in the presence of a catalyst at high temperatures.	26
	Deterioration at Elevated Temperatures	High temperature oxidation	Continuous layers of hard black magnetic iron oxide scale.	Internal—on fire row and superheater tubes; occurring in drums and headers. External—on surfaces in contact with furnace gases.	Internal—poor circulation resulting in overheating in a steam atmosphere. External—local flame impingement or poor circulation or presence of heat insulating scale on water side.	32 33
		Bursts	Longitudinal gaping split which may or may not be associated with ballooning.	Hottest side of generator or superheater tubes.	Water starvation, resulting in over-heating with corresponding reduction in mechanical properties of the tube until bursting occurs under working pressure.	34 35
		High temperature Creep cracking	Cracking and splitting associated with layers of high temperature oxide scale on tube surfaces.	In superheater tubes.	A combination of stress due to restricted expansion and overheating at temperatures around 550 deg. C.	36 37
	External Deterioration — General wastage by damp soot		Localized wasting of boiler metal which can only be seen after removal of soot deposits.	Wherever soot collects, especially where tubes enter lower drums and headers.	Presence of sulphuric acid formed by action of moisture (when shut down) on oxides of sulphur present in the oil fuel soot.	28 29

PITTING

Pitting is by far the worst type of corrosion encountered on the water side of marine boilers, both for the frequency of its occurrence and its intensely localized nature. Such corrosion occurs mainly in boiler water which has a pH value between 6 and 9 and which contains dissolved oxygen and salts.

A broad assessment of the rate and nature of corrosion occurring under boiler operating conditions on steel surfaces when in contact with boiler water over a range of pH values is

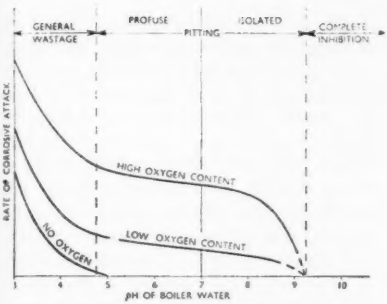


Fig. 2. Relationship between Corrosion of Steel and Condition of Boiler Water

shown in Fig. 2, from which it can be seen that isolated pitting may be anticipated even in waters with a *p*H value of 9.4. Pitting is also the main form of corrosion in slightly acid water, but the pits become more and more profuse with increasing acidity until they gradually merge, as general wastage.

The initiation and development of a pit on steel surfaces when in contact with boiler waters (in the range slightly acid to slightly alkaline) is directly associated with the instability of surface films. These films comprise a semi-porous cap of corrosion products which forms over areas being attacked, and which permits the free passage of ions but restricts the ingress of oxygen. Such corrosion can be described as a stage midway between general attack in acid waters and complete inhibition. It develops as a result of electrochemical action involving a difference in potential between existing surface coatings and patches of bare exposed metal, with sacrificial anodic attack of the latter. The rate of corrosive attack is proportional to the magnitude of current which flows, which in turn depends upon relative surface area of the anode and cathode, nature of the electrolyte, temperature, rate of supply of oxygen to depolarizing cathodic areas, and other factors. The characteristics of established pits, such as position, rate of development, nature and appearance of corrosion products, etc., depend upon the conditions existing in the various sections of a boiler, and the main types encountered in marine boilers may be classified under three headings:—

"Air bubble" pitting, which is that associated with oxygen bubbles adhering to the metal surfaces under stagnant conditions.

"Scab" pitting, which occurs in the hotter areas of generating surfaces, and is always associated with hard caps of corrosion products which have been described as "scabs."

"Soft scab" pitting in which the cap of corrosion products is somewhat softer and redder in nature. This type of pitting occurs mainly in drums, superheater tubes, headers, main steam pipes and on any such surfaces which are not submerged in water during idle periods.

Air Bubble Pitting. During idle periods when the boiler stands filled or partially filled with cold water, a substantial amount of air may find its way into the water. Solubility decreases rapidly with rise in temperature, and thus, consequent on small daily fluctuations in room temperature, conditions may result in air being thrown out of solution to collect as bubbles on the metal surface. These bubbles attach themselves to the adjacent metal surfaces throughout the entire boiler below the water level. At some areas, such as the roofs of water drums, etc., there may be a greater concentration of bubbles than elsewhere. Furthermore, when prolonged non-steaming periods are anticipated, it is the general practice to fill the idle boiler completely with water, and this extends the surfaces liable to suffer from this form of deterioration. Typical examples of air bubble pitting existing on a range of water surfaces are given in Figs. 3 and 4.

Where such bubble attachment has occurred, a potential difference will exist on the metal surface between the oxygen rich areas under the bubble and the surrounding areas in contact with water which will be less rich in oxygen.

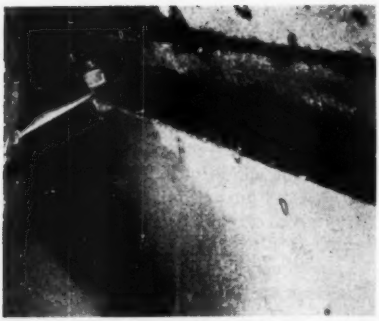
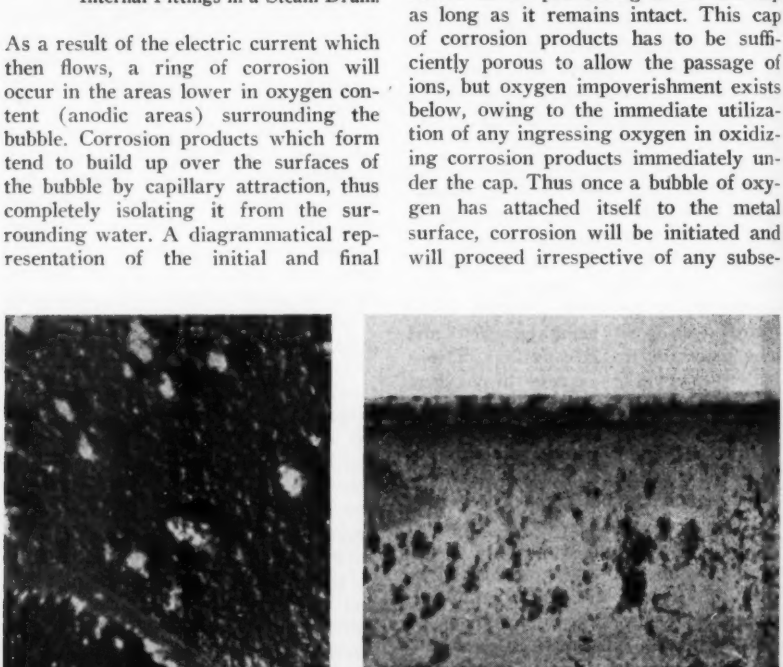


Fig. 3.—Soft Caps of Red Iron Oxide surmounting "Air Bubble" Pitting on Internal Fittings in a Steam Drum.

As a result of the electric current which then flows, a ring of corrosion will occur in the areas lower in oxygen content (anodic areas) surrounding the bubble. Corrosion products which form tend to build up over the surfaces of the bubble by capillary attraction, thus completely isolating it from the surrounding water. A diagrammatical representation of the initial and final stages in the development of this type of corrosion pitting is outlined in Fig. 5.

As the electrolytic reaction proceeds and the cap of corrosion product around the bubble thickens, oxygen inside the bubble is used up in converting the corrosion products into their fully oxidized state, leaving inert nitrogen inside the developing pit. Thus a stage is finally reached where the entire amount of oxygen under the cap of corrosion products is used up. The polarity of the small electric cell is then reversed, the area around the bubble becoming cathodic and that beneath the bubble anodic, corrosive attack proceeding below the cap as long as it remains intact. This cap of corrosion products has to be sufficiently porous to allow the passage of ions, but oxygen impoverishment exists below, owing to the immediate utilization of any ingressing oxygen in oxidizing corrosion products immediately under the cap. Thus once a bubble of oxygen has attached itself to the metal surface, corrosion will be initiated and will proceed irrespective of any subse-



(*a*) (*b*)

Fig. 4.—The Result of "Air Bubble" Pitting. (*a*) The Roof of a Steam Drum. (*b*) The Bore of a Tube.

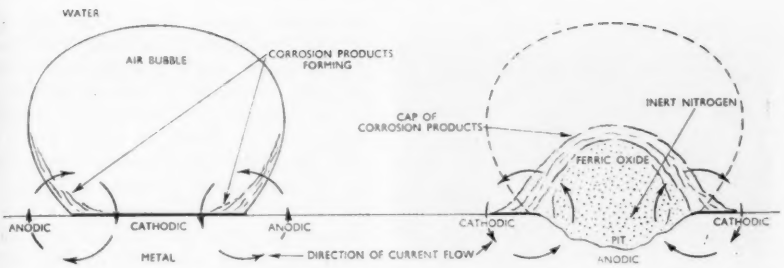


Fig. 5. Two Stages in the Formation of an Air Bubble Pit

quent changes, until the boiler is emptied and dried, causing the collapse of the cap of corrosion products, which will be found to consist of an outer layer of soft red iron oxide (Fe_2O_3), with an inner core of powdery black, magnetic iron oxide (ferroso-ferric oxide, Fe_3O_4) which fills the pit.

This form of deterioration is not so serious when it occurs on drums, owing to the thickness of metal and accessibility for cleaning. Where, however, it occurs on boiler tubes, it may be very serious and has been known to constitute up to 30 per cent penetration in one stagnant period of two weeks' duration. Moreover, it may initiate scab pitting on generator tubes if subsequent boiler water treatment is not employed. In some cases, examination suggests that the corrosion nodule is broken up by turbulence during subsequent steaming periods. Observations suggest that corrosion need not necessarily recommence over an existing broken nodule, but if possible it is best to brush away such corrosion products before refilling the boiler. The application of a layer of graphite on the drums appears to assist in immunizing from this form of attack, but this procedure is not to be advised on surfaces transmitting heat at higher rates owing to the possibility of localized breakdown with subsequent concentrated attack at the bare steel thus exposed. The only way to ensure complete freedom from this form of deterioration apart from completely emptying the plant and keeping it dry with the aid of heaters or dehumidifiers, is to exclude all oxygen from stagnant boilers. This may be achieved by adding a small amount of sodium sulphite (which will react with the remaining free oxygen) to the boiler water which is already treated with boiler compound.

"Scab" Pitting. A more virulent type of pitting may occur on the hotter areas of generating surfaces such as the hotter side of fire row tubes in a water-tube boiler. A hard, firmly adhering, bluish-black, scaly scab of corrosion products is formed over the developing pit which, once formed, may rapidly lead to the complete penetration of the tube wall. Examples of this form of deterioration are given in Figs. 6 and 7. A typical scab removed intact from a boiler tube is shown in Fig. 30, and comprises layers of hard, black, magnetic iron oxide in

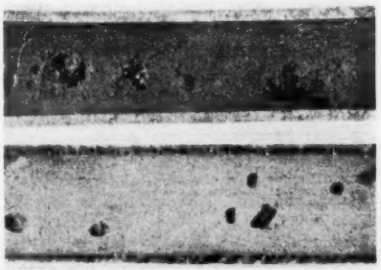


Fig. 6.—"Scab" Pitting on the Bore of a Generator Tube. In the lower illustration the hard scaly scabs have been removed by pickling to expose deep hemispherical pits.

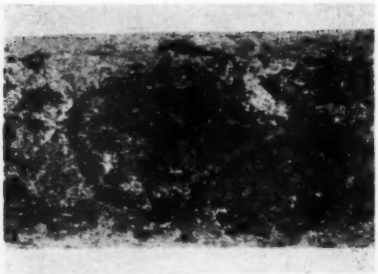


Fig. 7.—Massive "Scab" Pitting on the Water Side of a Fire Tube.

contact with the base of the pit, surmounted with a cap of red iron oxide which exudes beads of ferric chloride when dry.

The mechanism, and the precise conditions suitable for the initiation, of scab pitting are not, as yet, clearly understood. Initiation of this attack may involve oxygen bubbles, non-homogeneity of boiler tube steels, cracks in the oxide scale, or stress concentrations, but localized breakdown of semi-stable protective surface films is undoubtedly the predominating cause, which has already been shown to be a function of the composition of the boiler water. It has already been shown that below *p*H 6, protective films are completely unstable, and between *p*H 6 and 9.6 they are only partially stable, and it is in waters coming in this latter category that this form of deterioration most frequently presents itself. Scab pitting is also invariably associated with the use of boiler water contaminated with moderate amounts of dissolved salts, e.g. up to 140 parts per million, which hinder the formation of stable protective films and which greatly increase the conductivity of the water as an electrolyte. In marine practice, such contamination is due to chlorides from the sea-water which has found its way into the feed water, but similar scabbing has been noted in land practice as a result of contamination by other salts such as sulphates.

A probable mechanism of scab pitting involves the initiation of anodic attack on the metal surfaces by one of the factors noted above, which will then proceed favorably in the presence of the salt-contaminated boiler water. This is illustrated diagrammatically in Fig. 8. As the pit develops under the anodic surface, alkali is formed at the cathode (i.e. in areas surrounding the pit) as a result of the accumulation of hydroxyl (OH) ions. This alkali will tend to stabilize oxide films surrounding the pit, and by spreading laterally may stifle attack at any other incipient anodic point in the vicinity. Thus the distribution of centers of initial attack may be to alter with time, some persisting, and others disappearing, until the position is more or less stabilized, and well defined pits are established. Where the boiler water contains significant traces of sea-water,

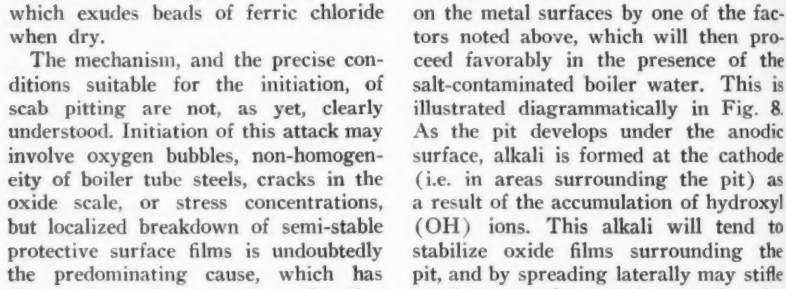


Fig. 8. Two Stages in the Development of a Scab Pit

both the anodic and cathodic products of electrolysis, ferrous chloride and sodium hydroxide respectively, are soluble, and will react to form an insoluble cap of ferrous hydroxide over the point of attack—a product identical with that noted in air bubble pitting. As hard scab pitting is always confined to areas of high heat transfer, it is evident that the hotter conditions prevailing cause the cap of corrosion products to compact and harden. The electrolytic cell set up under the scab will proceed to function so long as oxygen is available in the boiler water to depolarize the hydrogen diffusing outwards through the scab. As the scab thickens and hardens, so the pits become more established and localized, and more tenacious to the tube walls. The rate of attack may be increased with the higher temperatures involved, since the conductivity of the electrolyte increases with temperature. The result is an ever-deepening pit which has a non-selective penetration into the structure of the metal, as illustrated in Fig. 38. It is always filled with compacted layers of iron compounds in a reduced state, which are so hard as to defy all mechanical attempts at disruption by normal cleaning methods. Ultimately perforation takes place, either by leakage or splitting across the base of the pits as illustrated in Figs. 50 and 51.

It is conceivable that, in addition to the anodic attack, acid corrosion of the hydrogen evolution type may proceed under the scab, as a result of the formation of free hydrogen chloride by decomposition of any entrapped magnesium chloride at temperatures above 180 deg. C. (356 deg. F.).

In order to halt the attack when once started, it is necessary to remove the hard scab of corrosion products to expose bare metal in the pit, and then to re-form a protective film over the entire surface. Drying out the boiler may be of some limited value, and some scabs may flake away when tubes are subjected to thermal fluctuations with consequent expansion and contraction. The more usual tube cleaning operations, using power-driven wire brushes, fail to break away many scabs, and the only certain method is by acid pickling. This latter method is, however, fraught with many incidental problems, and if not carried out with laboratory precision may lead to even more virulent corrosive conditions, consequent on only partial removal of corrosion products and incomplete neutralization of residual acid.

Experience has indicated that scab pitting can be initiated during a relatively short period of boiler operation with significantly contaminated feed, and in the absence of a boiler water treatment to give the water a definite degree of residual alkalinity. This attack will continue during subsequent steaming periods even though the degree of contamination is extremely slight. However, by giving the boiler water a definite positive alkalinity and maintaining a *p*H value of 10.0-11.0, corrosive attack is completely inhibited by making surface films both stable and self-healing should rupture occur by mechanical or other means. The nature of such films depends to some extent on the composition of the boiler compound used for the chemical treatment, and are usually a mixture of ferroso-ferric oxide and metallic phosphates. Further, it has been observed that in many instances this treatment modifies the physical nature of the cap of corrosion products, ultimately causing it to disintegrate and fall away. If, when this occurs, the composition of the water is such as to promote the formation of stable protective films, corrosion will cease and complete freedom from it will be obtained.

"Soft Scab" Pitting. The visual appearance of this form of attack, as may be seen in Figs. 9 and 10, is reasonably similar to that discussed in the previous section, except that the cap of corrosion products is much softer and contains a higher proportion of red iron oxide (Fe_2O_3). It occurs mainly in superheater headers, superheater tubes, main steam piping, and on any steel com-

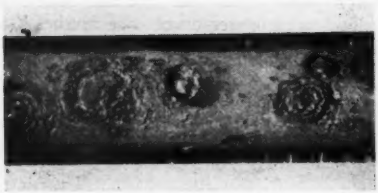


Fig. 9.—Large "Soft Scab" Pits on the Bore of a Superheater Tube.

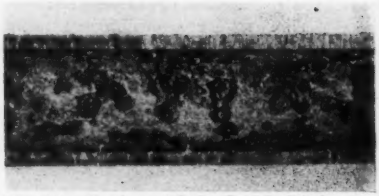


Fig. 10.—Profuse "Soft Scab" Pitting on the Bore of a Superheater Tube.

ponents above the water line in the boiler liable to come in contact with water which has primed from the boiler.

When such priming occurs (usually as a result of high total dissolved solids in the boiler water), the water deposited will evaporate, leaving salts on the surfaces concerned. Under humid conditions, such as will exist when the boiler is shut down, droplets of condensate may form on these metal surfaces and will then dissolve any salts in their immediate vicinity, producing small droplets of quite concentrated electrolyte. This will give rise to intense local attack at favorably disposed areas, the sequence of development being somewhat similar to that previously discussed. The initial and final stages in the development of a soft scab pit are illustrated diagrammatically in Fig. 11, from which it can be seen that electrolytic attack is initiated by a difference in potential between the inside and outside of the droplet consequent on an oxygen content gradient. Subsequently, corrosion products forming at an intermediate position produce a semi-permeable membrane and establish the corrosion cell. Final products of electrolysis comprise black ferroso-ferric oxide (Fe_3O_4) on the inside and fully oxidized ferric oxide (Fe_2O_3) on the outside. Subsequent steaming will dry out moisture from the corrosion cap and further oxidize the iron compounds present to the "red rust" (Fe_2O_3) condition; this is a frequent characteristic observed in superheater tubes where operation has been intermittent.

Once this type of deterioration has been established, it may recontinue whenever the boiler is shut down and condensation takes place. Droplets of salt-contaminated water primed from the boiler may reach areas some distance away, and attack of the type noted has been found on maneuvering valves, turbine blading, etc. In order to be free from this form of corrosion it is necessary either to fill with correctly treated water components liable to be affected, or, on running down the boiler, to ensure complete dryness by the provision of a form of dehumidification.

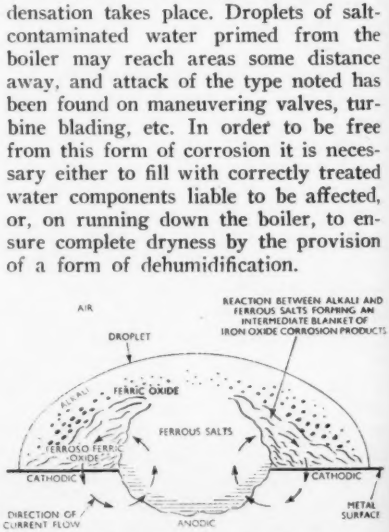


Fig. 11. Formation of a Soft Scab Pit

GENERAL WASTAGE

General internal wasting describes corrosive attack of a more uniform nature rather than selective local attack by pitting. Reduction in metal thickness occasionally may be evident over comparatively large areas, and the three most common forms of attack are: (1) Thinning of tube wall. (2) Necking and grooving by circumferential bands of wastage around tubes or stays in cylindrical boilers, where they enter boiler plates. (3) Galvanic attack by dissimilar

metals in contact or at surface irregularities.

These forms of attack are described and discussed in more detail in the following sections.

Thinning. When untreated boiler water is contaminated with sea-water, it tends to become slightly acid in nature (under boiler operating conditions) possibly as a result of the partial decomposition of the salts present, and in particular of magnesium chloride. Additionally, dissolved carbon dioxide results in acidity. The relationship between the *p*H of water, the oxygen content, and the type and rate of corrosion, has already been discussed, and has been shown diagrammatically in Fig. 2, from which it can be seen that below a *p*H value of 5.0, corrosive attack is no longer selective but general.

Such corrosion of steel in acid solutions proceeds in an even and non-selective manner, by virtue of the fact that the resulting corrosion products are soluble, or do not form obstructive or protective films at the metal surface. At *p*H values below 5.0 metal wastage proceeds in the absence of oxygen, and is of the hydrogen evolution type. The rate of attack depends on a number of factors, including the degree of acidity of the water, temperature, and the nature of the surface being attacked. Additionally, it is possible that local isolated pockets of acid water may exist under blankets of corrosion products or deposits as a result of the decomposition of entrapped magnesium chloride, which will then cause localized acid attack even when the bulk of the boiler water may be neutral or alkaline. The presence of dissolved oxygen in the boiler water increases both the rate and range in which acid corrosion will occur.

Consequent on the sporadic rupture of unstable surface films in water of a *p*H value between 5.0 and 6.0, sites of attack change before any point of anodic attack has proceeded long enough to develop a local pit. As a result of boiler water continually tending to become slightly alkaline by the continuous formation of ferrous hydroxide, conditions liable to result in this form of deterioration are seldom encountered, but on occasions, mainly due to fortuitous sea-water contamination, general even wastage has taken place. Such an example is illustrated in Figs. 12 and 13, an example of this form of deterioration occurring under heavy deposits is given in Fig. 14.

Necking and Grooving. This type of corrosion, which is a special case of general wastage, occurs mainly in cylindrical boilers on smoke tubes, combustion chamber stays, or longitudinal stays where they enter boiler end plates. A typical example of necking occurring

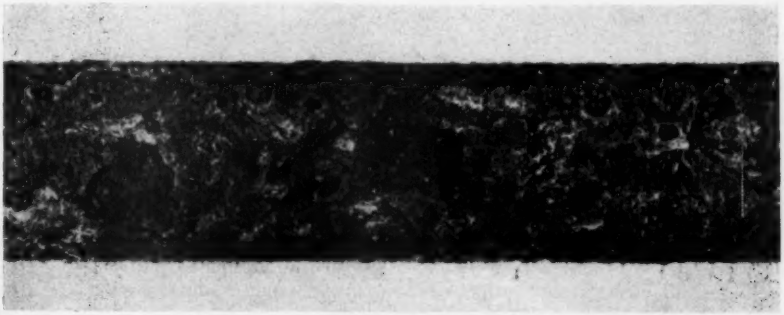


Fig. 12.—Blanket of Soft Loose Corrosion Products Surmounting "General Even Wastage" of the Bore.

Fig. 13.—Reduction in Wall Thickness by "General Wastage."

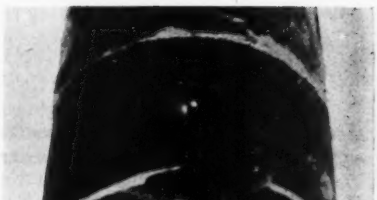


Fig. 14.—"General Even Wastage" Around a Tube End Under a Heavy Accumulation of Sludge.

on a fire tube where it enters the tube plate is shown in Fig. 15. Such wastage is confined to the end adjacent to the tube plates, and is undoubtedly due to the local flaking of protective films. At the tube ends, strains adequate to flake protective films on the tube surface are set up, consequent on expansion and contraction. Thus a circumferential band of bare metal is continually exposed to the boiler water. More or less intermittent attack will proceed at this area as the protective films successively break away, the amount of attack depending upon the relative corrosiveness of the boiler water.

Fig. 15.—"Necking and Grooving" Immediately Adjacent to where a Smoke Tube in a Cylindrical Boiler Enters an End Plate.

Additional to this continued mechanical exfoliation of films at tube ends, localized corrosion may be observed at areas coincidental with the junction of the tubes and end plates, where mill-scale on new unpickled tubes has been flaked away by the rolling and expanding operation when tubes were fitted in the boiler. On very rare occasions, attack in the capillary between tubes and rolling surfaces results from the concentration of salts, particularly magnesium chloride, and their subsequent partial decomposition to acid, leading to localized acid corrosion of the hydrogen evolution type. Such a case occurring at the edge of the rolling surface of a generator tube is illustrated in Fig. 16, in a water-tube boiler where it emerges into the steam drum.

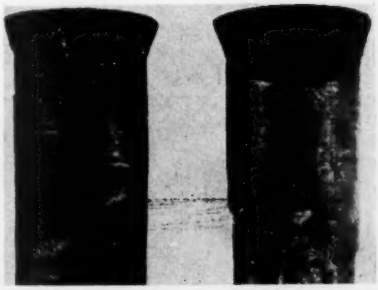


Fig. 16.—"Grooving" Around Water-Tube Ends where they Emerge from the Rolling Surface into the Drum.

As with thinning, wastage by "necking" and "grooving" is favored by the use of acid or nearly neutral boiler water. Such wastage is completely eliminated if the boiler water is given a correct alkalinity by chemical treatment.

Galvanic Attack. Dissimilar metals in contact in the presence of an electrolyte produce a short-circuited cell in which the less noble metal forms the anode, with consequent wastage, and the more noble metal forms the cathode, which is unattacked. The intensity of the attack depends upon the current density at the anode, which is a function of the conductivity of the electrolyte, the sizes of anode and cathode, polarization at the anode and cathode, and the potential difference between the two metals on an open circuit. Conductivity of boiler water is greatly increased by the presence of dissolved salts, and thus this form of deterioration will readily proceed in boiler water containing small quantities of sea-water. If, however, the boiler water is sufficiently alkaline, polarization—particularly at the anode—is greatly increased owing to the formation of stable protective films. The potential difference between various metals on an open circuit varies somewhat with the nature of the electrolyte in which they are immersed. In the case of the usual run of boiler waters, the following metals are set out in the order in which any metal is less noble (anodic) with respect to those which follow it: zinc, wrought iron, steel (cold worked), steel (normalized), brass, copper, gunmetal, monel metal, stainless steel, graphite.

Steel is thus anodic to a number of other nobler metals which may be used for fittings in boilers. In general, however, the area of steel is very large in comparison with that of the fittings, so that the current density at the steel is relatively small. Cases do occur where serious local galvanic attack is encountered, and an example is given in Fig. 17, which shows severe wastage in a tube end immediately adjacent to a stainless steel nozzle inserted inside a generator tube of a forced circulation boiler. In this instance, water treatment had been inconsistent, and it is appropriate to mention that subsequent laboratory experiments clearly demonstrated the value of a correctly maintained boiler water in inhibiting this galvanic attack.

Galvanic attack can be used in a broader sense to describe corrosion which will occur by electric currents flowing as a result of irregularities in the metal surface. Perhaps the best illustration of this is when mill-scale existing on the metal surfaces fractures to

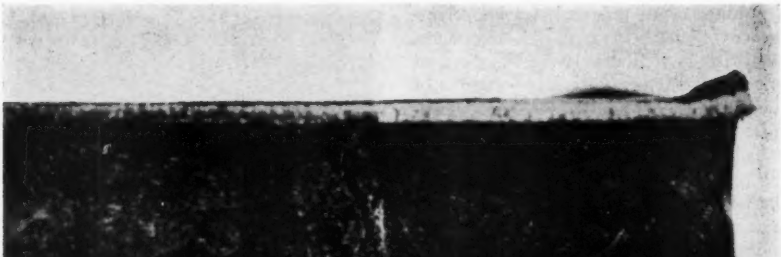


Fig. 17.—"Galvanic Attack" on the Bore of a Mild Steel Generator Tube Immediately Adjacent to where a Stainless Steel Nozzle had been Located in the Tube End.

expose bare metal underneath, the less noble bare steel dissolving, often locally with the formation of a pit. A good example of this is illustrated in Fig. 53, which shows lines of corrosion along Luder's lines on a tube wall, where strain has caused a thin surface film of iron oxide to fracture along planes of maximum stress. One particularly large pit has developed at the intersection of two of these lines of exposed metal. Other surface irregularities such as cold work, non-metallic inclusions, grease deposits, etc., each give rise to a potential difference, and the generation of local electrochemical activity with the commencement of this type of corrosive attack.

Cases are on record where galvanic attack on broken millscale surfaces of unpickled fire tubes, fitted in Scotch marine boilers, has proceeded with such rapidity that perforation has taken place in as short a space of time as two weeks. Quite often also, chains of pits have developed along the top surface of smoke tubes, as a result of galvanic attack under a layer of sludge which has been deposited between the tubes in zones where turbulence by steaming is at a minimum.

The principle of galvanic attack has also been used for protecting steel surfaces in boilers by sacrificial attack of less noble metal anodes, usually zinc slabs spaced throughout the boiler, but such protection has been found to be extremely local and it has been abandoned for more positive methods.

The foregoing refers principally to phenomena which take place at atmospheric temperatures and pressures, and hence which may occur when the boiler is cold. The significance of higher temperatures and pressures on galvanic attack are as yet less clearly understood, and it is probable that substantial modifications in the phenomena are involved. For example, it has recently been demonstrated that zinc may become cathodic to steel in certain waters at temperatures above 80 deg. C. (176 deg. F.).

CORROSION FATIGUE

The significance of a corrosive environment in substantially modifying the normal fatigue limit of a material in air has been clearly demonstrated by many previous investigators. In a corrosive environment a true fatigue limit is never attained because at any stress, however small, fracture will ultimately occur. For practical purposes, a corrosion fatigue limit may be specified for a definite number of stress applications in a given corrosive environment, and curves expressing the difference between the two phenomena for a range of stresses are given in Fig. 18.

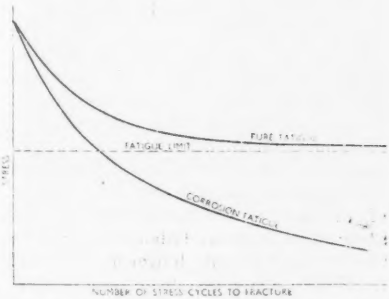


Fig. 18. Comparison between Fatigue and Corrosion Fatigue Curves for Mild Steel

Many boiler components are subjected to stress fluctuations during the operation of a boiler. In some cases these fluctuations may be frequent and severe, as in a generator tube where circulation may be defective. In other cases the fluctuations may be relatively infrequent and occur only when the boiler is heated up and cooled down. The influence of a corrosive environment (e.g. untreated boiler water) on the initiation and subsequent propagation of fatigue cracks involves the following possibilities:—

1. The surface of the metal may be roughened by corrosive attack and so give rise to stress concentration effects which have a significant adverse effect on the fatigue limit.
2. An extending crack will result in the continued breakdown of protective films at its terminus and thus leave bare metal continuously open to further corrosive attack.
3. More highly stressed areas are anodic to less highly stressed areas and attack is thus favored at the former. The terminus of a pit or crack coincides with a stress concentration, and hence maximum corrosive attack is encouraged at such points.

Corrosion fatigue fissures are characterized by the direct path which they take across the crystal structure of the material. They usually take the shortest path through the component being stressed, and show no preference to pass around crystal boundaries. The broadness and shape of the fissures, and the rate of crack propagation, depend upon the magnitude and frequency of the stress application and the nature of the corrosive environment. For instance, under mildly corrosive conditions but with high frequency and a stress concentration of fair magnitude, fissures may develop as hairline cracks. On the other hand, with a low stress concentration and low frequency, a corrosion fatigue fissure may be so broad and filled with corrosion products that it more resembles a pit. Figs. 40 and 41, show typical corrosion fatigue fissures frequently found in boiler tubes, where both the frequency and the magnitude of stress concentration are comparatively high. These can be compared with Fig. 39, which shows a corrosion fatigue fissure in a main steam pipe, where the stress frequency has been so small that the deterioration may be more aptly described as stress corrosion, which is a form of preferential corrosive attack encouraged by a potential difference existing between stressed and unstressed areas.

Corrosion Fatigue in Tubes. Corrosion fatigue cracking may be found in almost all boiler components, but is most common in boiler and superheater tubes.

In boiler tubes, the stress frequency can be attributed to rapid changes in surface temperature along the bores, as a result of intermittent circulation. The cause of poor circulation in natural circulation boilers is a much discussed phenomenon, and involves correct distribution of heat in the furnace through the tube bank, a complete separation of steam and water in the steam drum, and other factors involving the actual design of the boiler, which are beyond the scope of this paper. Laboratory experiments have shown, however, that intermittent slugs of steam and water, passing along the boiler tube, will result in pulsating thermal skin stresses of sufficient magnitude to result in a network of tiny fatigue cracks developing on the bores.

The rate of propagation of these cracks is usually slow, but where boiler tubes operating under these conditions are under the influence of additional tensile stresses the rate of crack development is correspondingly increased. Thus, tubes which are curved or buckled as a result of local restriction to expansion are the first to exhibit this form of deterioration, as a result of bending moments inducing skin tensile stresses along the bores. For this same reason, failures are often immediately adjacent to rolling surfaces where tube movement has induced tensile stresses of considerable magnitude. This form of deterioration is not entirely confined to generator tubes, as some boiler designs are such that circulation in some of the downcomers (usually the first few rows) is extremely sluggish and may, due to violent changes in steaming rate, result in these latter tubes alternating as risers and downcomers. Examples of the profuse nature of cracking which may be found on generator tubes under these conditions is shown in Fig. 19.

The influence exerted by the nature of the generating surface on the production of steam bubbles is well known, and a generating surface corrugated by a large number of tiny cracks offers many nuclei for steam bubble genera-

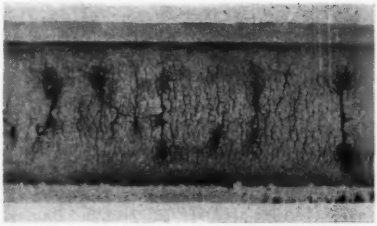


Fig. 19.—Network of "Corrosion Fatigue" Fissures on the Bore of a Generator Tube. Main Fissures are Circumferential, in a Network of Crazy Cracking.

ing surface offered by fissures, may re-
tion. This, together with the large heat-
sult in violent liberation of steam bub-
scribed as "crazy cracking."
bles, which will then add further to the bad circulation, and be a contributory cause to the development of corrosion fatigue cracks.

The orientation of cracks over a tube surface is governed by the resultant of the stresses involved, to which the stresses imposed by the normal working pressure of the boiler will contribute. Thus, cracking may be circumferential, oblique, longitudinal or may occur at random in all directions. Inherent surface markings on the tube wall such as scratches, over-rolling scores, pits, etc., often act as stress raisers, and influence the position and rate of cracking. Illustrations of this are given in Figs. 20 and 21, and incipient cracks radiating from the root of a manufacturing defect are revealed in the micro-section, illustrated in Fig. 54. When the cracking is completely at random it is described as "crazy cracking."

Fig. 20.—"Corrosion Fatigue" Cracks Developing from Spiral Scores along the Bore of a Generator Tube.

Fig. 21.—"Corrosion Fatigue" Fissures Radiating from the Base of Pits.

Fig. 22.—"Corrosion Fatigue" Fissures Between Tube Holes in a Steam Drum, Revealed after Pickling the Section.

A full understanding of the conditions resulting in the development of specific forms of cracking is useful to designers in indicating conditions existing in a particular boiler, and frequently metallurgical evidence has contributed in this manner to boiler design.

Corrosion Fatigue in Boiler Drum. Another source of corrosion fatigue is to be found in tube hole arborings of steam and water drums, where static stresses may be inherent from the manufacturing operation and may be supplemented to a small degree by stresses imposed during steaming. However, such deterioration is usually coupled with deep selective pitting, and is not very frequent in its occurrence. An example of such cracking in the arborings is given in Figs. 22 and 23, but it must be noticed in this case that the tubes were not entering the drum radially, and the combination of the original machining operation in the arborings, cold work during tubing operations, and corrosive conditions during operation, all contributed to this failure.

Corrosion Fatigue in Main Steam Lines. Steam lines also exhibit corrosion fatigue cracking. An example is shown in Fig. 24, which illustrates a fissure in the base of a corrugation. In this case it is possible that original surface roughening exerted an influence on the subsequent development of corrosion fatigue fissures. Frequency of stress application in this component was extremely small, and the fissures, one of which is illustrated in the photomicrograph Fig. 39, had only penetrated to a small depth.

Many other examples of corrosion fatigue cracking have been reported from a wide range of both marine and

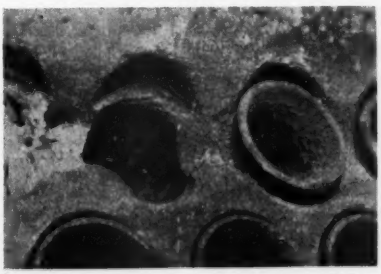


Fig. 23.—Progressive Weakening of the Steam Drum by Repeated Attempts to Grind Away Developing Cracks.

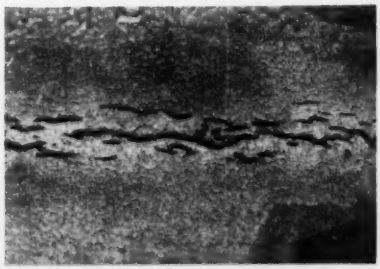


Fig. 24.—"Corrosion Fatigue" Fissures in the Base of a Corrugation in a Main Steam Pipe.

land installations. For instance, desuperheaters are especially prone to this form of deterioration, as may be expected from the conditions under which they operate. Boiler components other than those enumerated frequently fail by corrosion fatigue cracking, and an individual examination of each case usually brings to light conditions which have existed and which have accounted for this form of deterioration.

Reduction to a minimum of corrosive conditions by water treatment will, of course, minimize this form of deterioration, and indeed may, if the stresses imposed are less than the fatigue limit of the material, result in its complete elimination.

STRAIN AGE EMBRITTLEMENT

Occasionally, boiler components have failed by spontaneous cracking, as a result of shock. Rivet heads are perhaps most frequently affected in this manner, although boiler tubes have been known to fracture immediately above or below a drum during a cleaning or a detubing operation. An instance of the latter is illustrated in Fig. 25. This form of deterioration has been termed "strain age embrittlement," and is directly connected with the reduction of impact properties of certain mild steels following cold work and aging. The joint effect is to reduce the impact properties of the material to a very low value at room temperatures. At boiler operating temperatures, however, the impact properties of such strain aged material are substantially higher, and thus failure seldom takes place under operating conditions. Surface irregularities offered by sharp pits, or grooves caused during the tube rolling operation, offer favorable nuclei for the initiation of spontaneous rupture of the brittle steel.

Critical examination of a failed component frequently reveals conditions which have been responsible for inducing the cold work, as for example, the case of a failed tube end (Fig. 25,) which had been cold-worked by overrolling, followed by repeated bending

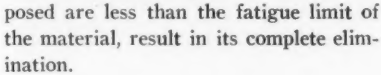


Fig. 25.—Fissure in a Generator Tube as a Result of Spontaneous Fracture (Following a Mild Shock) on "Strain Aged" Material.

moments imposed immediately below the tube plate by tube movement during steaming periods. A small specimen removed from the affected area proved that the impact properties in this zone were very low at room temperatures.

Repeated tubing and detubing operations may induce cold work in the arborings of tube plates, and although the stresses may be insufficient to result in spontaneous failing by cracking, they frequently encourage stress corrosion cracking. Annealing must be carried out in order completely to restore impact properties in the boiler component, but unfortunately strain age embrittlement gives no indication of its presence until spontaneous cracking has occurred.

CAUSTIC EMBRITTLEMENT

Although considerable attention has been devoted by a number of investigators to the causes of this form of deterioration, and opinions still remain divided, its appearance in marine boilers employing distillate feed is fortunately extremely rare. When it does occur, its presence is more likely to be found in such components as boiler plates, riveted seams, and under rivet heads, than in boiler and superheater tubes. It is more common in boilers using chemically treated raw water, where conditions for its development are more likely to be found. Caustic embrittlement results from a combination of causes involving quality of steel, non-uniform static stresses, elevated temperature, a high concentration of caustic soda, and possibly the presence of a catalyst such as sodium silicate. There are several schools of thought as to the relative importance of these influences.

Certain conditions of boiler water composition result in free sodium hydroxide being formed and concentrated in seams and crannies, mainly as a result of the combined influence of capillary attraction and evaporation. Under the influence of temperature, concentrated caustic soda attacks stressed steel in an intergranular manner, possibly by hydrogen absorption. The rate of attack is accelerated with increase in temperature and stress concentration. An example of a fissure which developed as a result of caustic embrittlement is illustrated in Fig. 26, which shows a crack between riveted holes on a wrapper plate. Cracks associated with this form of deterioration progress through the steel in an intercrystalline manner, an example of which is illustrated in Fig. 43. This intercrystalline characteristic (in the absence of evidence of overheating above 500 deg. C. or 932 deg. F.) always distinguishes this form of deterioration from corrosion fatigue

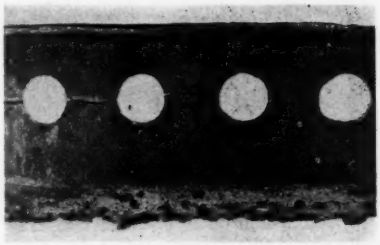


Fig. 26.—"Caustic Embrittlement" Between Rivet Holes on a Wrapper Plate.

cracking, which has been discussed in a preceding section. Where alternating stresses are also present, corrosion fatigue fissures and caustic embrittlement may combine to give failure, which would then be a combination of inter- and trans-crystalline cracks. Alternatively, the rate of caustic embrittlement cracking may be increased, but in this case it would appear that alternating stresses are of much less importance than a combined chemical action and static stress.

It has been suggested that sodium silicate must be present as a catalyst before caustic embrittlement occurs, and work is still proceeding to establish the exact causes of this type of boiler failure and to formulate inhibitive measures to prevent it. So far, however, it has not been possible consistently to reproduce results. It has been claimed that this form of deterioration does not occur if a definite sulphate/hydroxide ratio is maintained. The benefit of certain organic compounds for this purpose is also under investigation. Many data have already been published on the subject, and it is not necessary to mention them in any greater detail here. "Caustic embrittlement" is perhaps the most widely discussed form of boiler deterioration, but in point of fact remains by far the most infrequent encountered in boilers using distillate feed.

DETERIORATION AT ELEVATED TEMPERATURES

High Temperature Oxidation. In a general sense all types of corrosion which have been described in the preceding sections are a form of oxidation, since iron is converted into its oxide when corrosion takes place. The term "oxidation" can, however, be used in a more restricted sense, referring to the direct reaction between gaseous oxygen and iron which takes place at temperatures above 550 deg. C. (1022 deg. F.). This form of oxidation may thus be more aptly described as "high temperature oxidation." The reaction which takes place when iron is heated in a current of steam to produce ferrosoferric oxide is so well known that it needs no description here, but the influence of time and temperature on the rate of the reaction plays an important part in the deterioration of boiler components by reduction of tube wall thickness. Under normal conditions of heat transfer in generator and superheater tubes, temperatures are insufficiently high for direct oxidation to take place, but occasionally, as a result of fortuitous circumstances, excessive tube wall temperatures occur with consequent deterioration. Attendant on high temperature oxidation of tube walls are other forms of deterioration, such as bursting and the development of high temperature creep cracks. These latter phenomena are discussed in the following sections.

The nature of the scale produced on generator surfaces by overheating often supplies evidence as to the conditions existing at the time of deterioration, and thus an understanding of the nature and rate of scale formation on steel surfaces, over a range of elevated temperatures, is valuable to the engineer. This subject has received attention by many workers, and a great deal of published matter is available. Curves expressing the relationship between the rate of scale formation and time for a range of temperatures have been obtained experimentally on sections of boiler tube steel

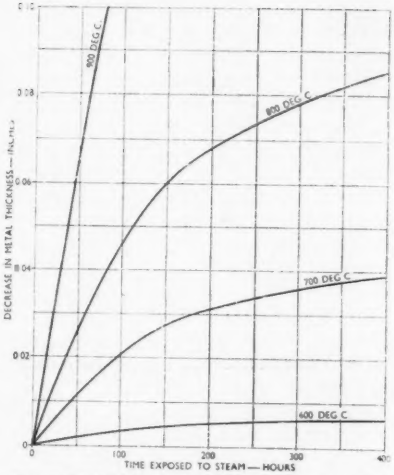


Fig. 27. Relationship between Time and Decrease in Metal Thickness for Mild Steel in an Atmosphere of Superheated Steam at Various Temperatures

and are given in Fig. 27. Another phenomenon which gives evidence of the conditions responsible for the development of high temperature iron oxide scale is the crystalline structure which is frequently exhibited on the outer surface. It has been observed that higher temperatures promote the growth of correspondingly larger surface crystals, and a visual examination often gives an indication of the temperatures which have been involved.

High temperature scaling may occur in any parts of a boiler subjected to heat transfer, e.g. in generator tubes where water flow has been partially restricted. Some workers have shown that the rate of oxidation is slightly greater in air and carbon dioxide than in steam; thus it is often found that the layer of scale on the outside of the tube surfaces is thicker than its corresponding layer on the bore, even when taking into account the fact that the outside surface temperature is a few degrees higher than the temperature of the bore. High temperature iron oxide scale is hard and black, and often appears as a series

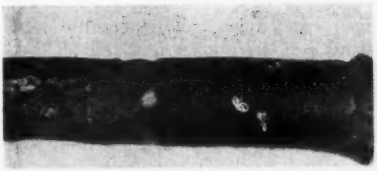


Fig. 28.—External Wastage of a Boiler Tube Immediately Above the Water Drum Rolling Surface by "Damp Soot Deposit Attack."

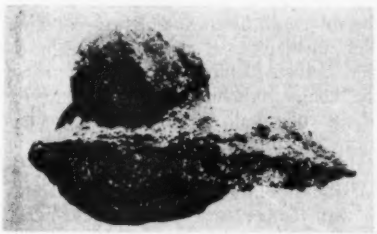


Fig. 30.—Enlarged View of a Complete Scab Removed from a Scab Pit on a Generator Tube.

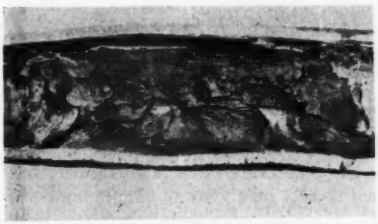


Fig. 32.—Layers of High Temperature Iron Oxide Scale.

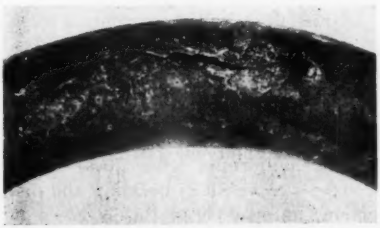


Fig. 33.—Reduction in Wall Thickness to the Point of Perforation Around a Superheater Bend by High Temperature Oxidation.

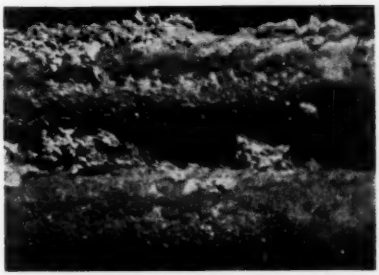


Fig. 29.—Accumulation of Soot and Ash from Burnt Oil Fuel on Superheater Tubes.

Fig. 31.—Thick Red Insulating Blanket of "Hard Water" Scale which has Built Up on the Water Surface of a Fire Tube, and which has been Fractured to Expose Tube Below.

of scaly layers, but might equally exist as one thick layer, depending upon conditions existing during its development.

Where heat insulating scales from the feed water are allowed to build up on the steam generating surfaces, such as illustrated in Fig. 31, they stifle internal oxidation, but, by reducing the rate of heat transfer and thus increasing wall temperature, they encourage overheating and high temperature scaling of the outside walls in contact with furnace gases. Occasionally, due to general water starvation throughout the entire boiler, the whole of a boiler may in some small degree be affected by high temperature oxidation, and indeed, in one case brought to the authors' attention, water starvation was so complete that the iron steam reaction proceeded so vigorously that the reaction became exothermic.

Only after the whole incandescent area was copiously flooded with water, was the reaction stifled. Often, the presence of high temperature oxide scale in a particular zone gives some indication of the heat distribution and water circulation in the boiler, and thus influences future modifications and improvements.

High temperature oxidation is found most frequently in superheater tubes, which operate at higher temperatures than boiler tubes and therefore are more susceptible to overheating, especially where steam flow has been sluggish. Frequently, a gradual increase in scale thickness from the inlet to the outlet end of a superheater tube indicates a progressive rise in tube wall temperature. On the other hand, local flame impingement, such as may occur around bends, gives a local area of high temperature scaling which may remain undetected until perforation occurs. An example of this is illustrated in Figs. 32 and 33.

It must be remembered that once overheating of a heating surface has taken place and a layer of scale has been formed, the rate of heat transfer across the section is reduced, with a corresponding rise in tube wall temperature, so that oxidation and scaling proceed more rapidly than ever. Another factor which influences the rate of scale formation is the expansion of the layer of scale which accompanies oxidation, and the difference between its coefficient of linear expansion and that of the parent metal. This latter point, together with thermal stress liable to occur along the wall of a tube, and especially around bends, may cause layers of scale to flake away, and thus continually expose fresh overheated metal to oxidizing atmospheres with a consquent increased rate of deterioration. An example of this is illustrated in Fig. 32. Examination of the micro-structure of an oxidized tube frequently shows marked decarburization of the surfaces similar to that illustrated in Fig. 45.

The precise assessment of boiler and superheater tube wall temperatures existing in a fired boiler is a difficult matter, and several methods are available to the engineer. The attachment of thermocouples to outside tube walls by "Jubilee" clips, brazing, peening, and other such methods have been found to give high results, mainly due to conduction back along the overheated leads to the hot junction. A new method, at present being developed, consists of passing the leads of a thermocouple through the tube wall by means of heat-resisting plug insulators. A composite hot junction is then formed by the two arms of the couple touching the inner surface of the tube wall opposite to the insulators. This obviates inaccuracies due to conduction back along the leads to the hot junction, and detects small changes in wall temperature on the steam generating surfaces.

While it is appreciated that this form of pyrometer would not be easy to install in a large number of representative points throughout a boiler, it could be used as a standard to assess the inaccuracy of simpler but more practical forms of tube wall temperature measurement over a wide range of operating conditions, which could then be disposed throughout the whole boiler. Up to the present, however, the engineer has had to rely on the more simple methods, the accuracy of which has often been questioned by the appearance of such forms of deterioration as high temperature scaling.

Distortion of boiler components often occurs alongside high temperature oxidation, but on the other hand this may develop at temperatures insufficiently high to cause scaling. In tubes, it usually results from the tube bank not expanding uniformly, when local strain or creep may result in buckling and distortion; an example of this is given in Fig. 52. While this form of deterioration is not in itself serious, it frequently plays an important part in the failure

of tubes as a result of corrosion fatigue cracking. This arises from the fact that once a buckle has been formed, any further strain induces bending moments around the buckles with high tensile skin stresses on one side of the tube. This greatly accelerates the development of circumferential corrosion fatigue cracks, or, if temperatures are sufficiently high, causes exfoliation of developing layers of high temperature scale and the exposure of bare metal to oxidizing influences.

Some forms of superheater tube, especially those which have supporting lugs between the saturated and superheated legs, frequently distort by curling, due to differential expansion which may not be fully accounted for in the design. As will be shown in later paragraphs, such distortion in superheater tubes frequently plays an important part in the development of another form of deterioration termed "high temperature creep cracking."

Bursts. When bursting occurs in boiler tubes, it may, almost without exception, be traced to local or general water starvation, which has resulted in a rise in tube wall temperature. Causes of such starvation are many, and include abnormally low water levels, irregular distribution of hot gases in the furnaces causing faulty circulation, bad design, local flame impingement, foreign bodies left in the water drums, violent fluctuations in steaming rate and many other operational peculiarities which are outside the scope of this paper. Characteristics of the bursts often give an indication of conditions which have existed in the tube immediately prior to its failure. These include the actual dimensions of the burst, nature of scale surrounding the fissure, and the microstructure of the material at the edge of the fissure and in adjacent zones. In some cases a rise in tube wall temperature is consequent on a build up of heat insulating scale from impurities in the feed water, or an accumulation of corrosion product sludge, often loosely bound together by oil or grease.

The actual mechanism of a burst is approximately as follows: water or steam starvation results in a progressive rise in tube wall temperature with a corresponding reduction in mechanical properties. A point is eventually reached where stresses induced in the tube wall by the working pressure of the boiler exceed the elastic limit of the material, and bursting ultimately occurs. Where the rate of overheating is relatively slow, or is maintained at temperatures around 700 deg. C. (1292 deg. F.), considerable swelling of the tube takes place prior to the ultimate fracture, and this form of burst is characterized by the ballooned appearance and the feathered edges of the split such as is illustrated in Fig. 34. Where, however, rises in temperature have been sudden up to abnormally high temperatures (900-1000 deg. C.) (1652-1832 deg. F.) ballooning does not have time to take place before catastrophic splitting occurs. Such a case is shown in Fig. 35.

Frequently, microscopical examination reveals the edges of a fracture to have a martensitic structure, thus giving a positive indication that tube wall temperatures were in excess of 900 deg. C. (1652 deg. F.) prior to the burst, which had been followed with drastic quenching of the edge of the fracture by the escaping water and steam. This sequence will retain a characteristic martensitic structure even in mild steel. A typical example of a coarse martensitic structure thus produced at the edge of a burst in a boiler tube is illustrated in Fig. 48. Micro-examination of adjacent areas will often give evidence as to the extent of overheating along a particular tube, but frequently it is necessary to examine the whole tube bank before the true cause of the overheating is determined. Some intermediate structures often found adjacent to bursts are illustrated in Figs. 46, 47, and 49, which should be compared with the micro-

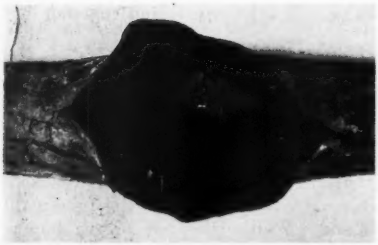


Fig. 34.—Burst in a Generator Tube Subsequent to Prolonged Wall Temperature of about 700 deg. C. (1292 deg. F.).

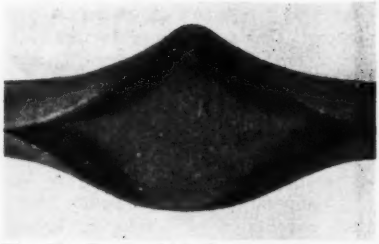


Fig. 35.—Burst which Occurred Spontaneously as a Result of a Sudden Rise in Boiler Tube Wall Temperature up to 950 deg. C. (1742 deg. F.).

Fig. 36.—Failure by "High Temperature Creep" Cracking Around a Superheater Bend.

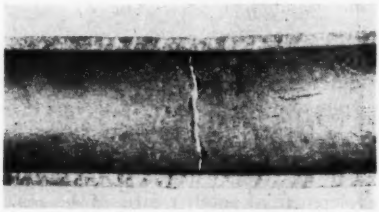
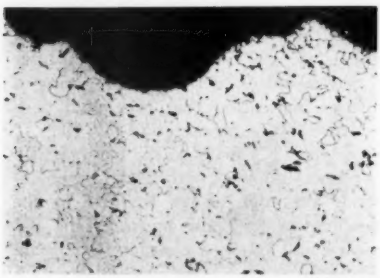
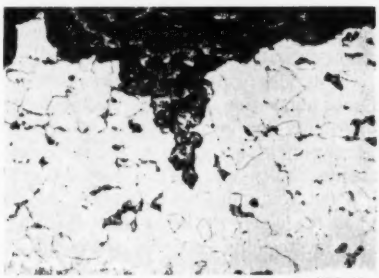


Fig. 37.—Perforation of a Superheater Tube in the Center of a Buckle as a Result of "High Temperature Creep."



× 100 dia.

Fig. 38.—Photomicrograph Illustrating the Normal Non-selective Penetration of a Corrosion Pit into the Surface of a Boiler Tube.

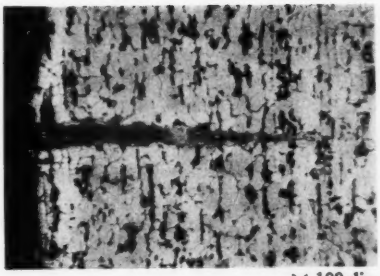


× 200 dia.

Fig. 39.—Broad Transcrystalline Corrosion Fatigue Fissure Filled with Corrosion Products, Resulting from Low Frequency "Corrosion Fatigue."

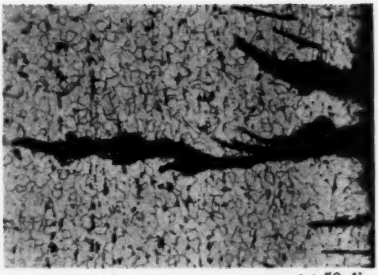
structure of average quality boiler tube steel illustrated in Fig. 44.

Distortion of ferrite and pearlite microstructure in the immediate vicinity of the burst, such as is illustrated in Fig. 55, frequently indicates failure after plastic deformation following prolonged elevated wall temperatures, but below 700 deg. C. (1292 deg. F.), which is the lower recrystallization temperature for mild steel. Where the tube has been at an elevated temperature for considerable periods prior to the burst, ultimate failure may be assisted by gradual weakening of the tube walls by the formation of high temperature scale, which in itself



× 100 dia.

Fig. 40.—Transcrystalline "Corrosion Fatigue" Fissure resulting from High Frequency "Corrosion Fatigue."



× 50 dia.

Fig. 41.—Typical Transcrystalline "Corrosion Fatigue" Fissures in a Boiler Tube Wall.

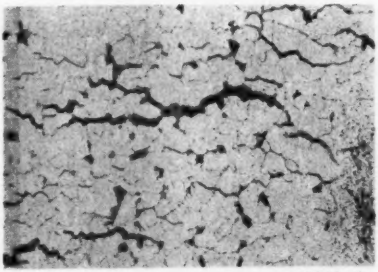


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Fig. 42.—Intercrystalline "High Temperature Creep Cracks" in a Superheater Tube Wall. Secondary cracking has taken place to the point of intercrystalline disintegration

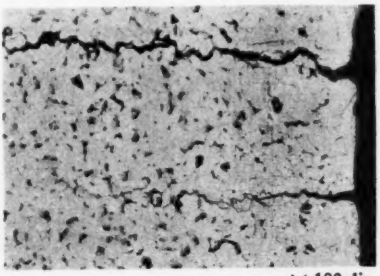

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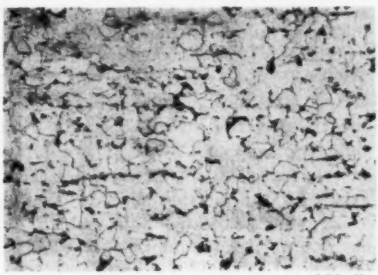
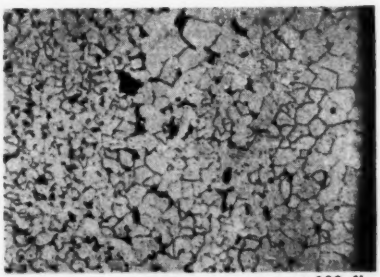
Fig. 43.—Intercrystalline "Caustic Embrittlement" Fissures Penetrating a Boiler Plate.



× 100 dia.

Fig. 44.—Typical Microstructure of Low Carbon Boiler Tube Steel.



× 100 dia.

Fig. 45.—Heavy Decarburization at the Surface of an Overheated Boiler Tube.

is a bad thermal conductor, and thus promotes even higher tube wall temperature.

Splitting of Superheater Tubes by "High Temperature Creep Cracking." While bursting is consequent on overheating of boiler tubes, a similar type of failure encountered at high temperatures in superheater tubes occurs mainly as splitting. Cracking may follow overheat-

ing, with corresponding reduction in mechanical properties, to the point of swelling and failure by splitting under the normal working pressure of the boiler, but more frequently it terminates a period of plastic deformation under the influence of stress and temperature, a phenomenon known as "creep." High temperature creep cracking is a form of intergranular weakness which is liable to be encountered in certain steels including low carbon steel. The temperature at which this cracking takes place in a particular steel is termed its equicohesive temperature, and for mild steel is about 550 deg. C. (1022 deg. F.). At this temperature, and depending upon other inherent characteristics, such as grain size, composition of the metal, etc., the grain boundary material of the structure becomes weaker than the aggregate, and thus, if in a state of stress, it fails by intergranular disintegration as is illustrated in Fig. 42.

Although these cracks are similar in appearance to those formed by caustic embrittlement, they can be identified from them by virtue of the temperatures involved, which result in the development of layers of high temperature iron oxide scale both on the corresponding metal surfaces and within the crack themselves. This form of deterioration has frequently been found where a difference in expansion has existed between the saturated and superheated legs of a superheater element. If such a stress concentration coincides with overheating, high temperature creep cracks develop, and ultimately constitute a penetration of the tube wall. Examples of this form of deterioration are illustrated in Figs. 36 and 37.

General Wastage by Damp Soot. Accumulation of heavy deposits from the oil fuel on the fire side of boiler components are a common occurrence, and their effect in reducing boiler efficiency is well known. The nature and rate at which these deposits collect, and their preference for particular zones, depend very largely upon the quality of the fuel and the design of the boiler. So much attention has been, and still is being, paid to this subject with regard to fuel economy, that consideration need only be given here to the effect that these deposits have on the deterioration of metallic surfaces with which they are in contact.

Deposits on generator and superheater tubes have a shielding effect from the furnace gases, and thus in no way encourage high temperature oxidation, but severe wastage is often found on steel surfaces beneath such deposits when these are allowed to remain and become damp during the periods when the boiler is not steaming. Consequently, steel surfaces most likely to suffer from this form of deterioration are those which are not easily accessible for cleaning purposes. Typical examples may be found where water tubes enter water drums, and superheater tubes enter superheater headers, and an example of a boiler tube which had been thinned to the point of perforation immediately above where it entered a water drum, by this form of deterioration, is illustrated in Fig. 28.

The most virulent forms of oil fuel deposits encountered in marine boilers are agglomerations of metallic oxides, sulphates, and chlorides, loosely bound together with carbonaceous matter. They are a mixture of green, white, yellow, and black salts, and are frequently partially fused together. Free sulphuric acid may also be present in varying amounts up to two per cent. Fuel oils from certain sources are rich in organic vanadium salts which, on burning, form vanadium pentoxide, having a low fusing temperature. This oxide exerts considerable influence on the physical properties of other components of the deposits, forming a eutectic of comparatively low melting point which runs over the tube as a slag and is very difficult to remove.

The presence of corrosive oil fuel deposits can be explained as follows:—

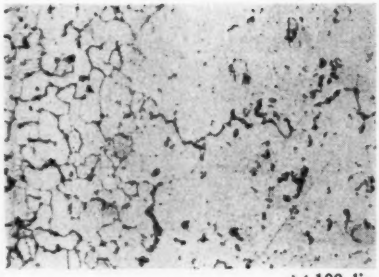


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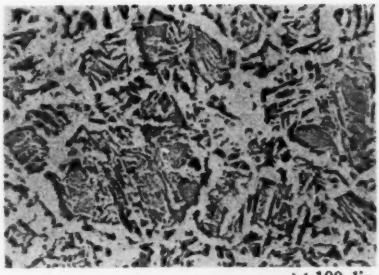
Fig. 46.—Spheroidization of the Pearlite in a Superheater Tube Wall as a Result of Prolonged Heating at Temperatures just below 700 deg. C. (1292 deg. F.).



× 100 dia.

Fig. 47.—Coarse Widmanstatten Structure Subsequent on Overheating Followed by Fairly Rapid Cooling.

× 100 dia.

Fig. 48.—Coarse Acicular Martensite Retained in a Low Carbon Steel Boiler Tube at the Edge of a Burst by Drastic Quenching from around 950 deg. C. (1742 deg. F.)

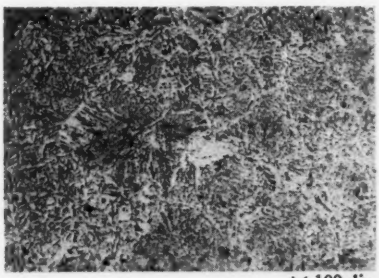


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Fig. 49.—Troosto-Sorbite in Boiler Tube Steel Formed by Quenching between 750-850 deg. C. (1382-1562 deg. F.).

Fuel oils may contain up to 1.5 per cent of sulphur which, when burnt, forms oxides of sulphur, some of which may be thrown down with the ash (mainly metallic oxides) and some unburnt hydrocarbons. Perhaps the main source of ash in marine boilers is sea-water, which frequently finds its way into the fuel oil with which it forms an emulsion, thereby depositing large quantities of metallic sulphates and chlorides in the furnace.

Ash then tends to collect in stagnant corners or pockets in the boiler, particularly where there is reduced gas flow, such as between tubes and water drums towards the back of the tube bank, re-entrant corners, and blind passages. An example of such an accumulation on superheater tubes is shown in Fig. 29. While the boiler is steaming, these accumulations are dry and inactive, but when the boiler is shut down, they absorb moisture as a result of the hygroscopic nature of some of the constituents. Dripping boiler components may substantially increase the moisture content, and it is not unusual to find free sulphurous or sulphuric acid being formed, which then trickles through deposits and dissolves metal with which it comes in contact by normal acid corrosion of the hydrogen evolution type. Some free sulphuric acid will combine with the metallic oxides in the deposits to form a variety of metallic sulphates, e.g. sodium sulphate, nickel sulphate, calcium sulphate, iron sulphate, etc. These metallic sulphates are usually

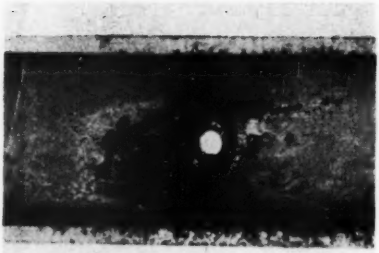

Fig. 50.—"Scab" Pitting which has Developed in a Boiler Tube Wall to the Point of Perforation.

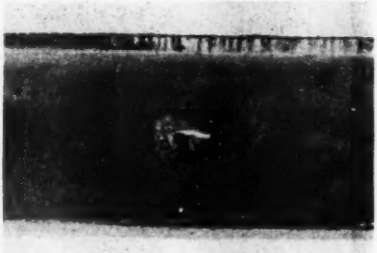

Fig. 51.—"Scab" Pit in a Boiler Tube Wall Constituting a Weak Zone which Ultimately Failed by Splitting at Elevated Temperatures.

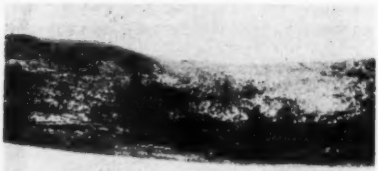

Fig. 52.—Local Bulging and Distortion along a Boiler Tube Wall Resulting from Restriction to Expansion.

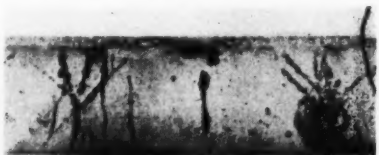

Fig. 53.—Lines of Corrosion where Planes of Maximum Compound Stress have Fractured Normally Protective Surface Films to Initiate Galvanic Attack at the Less Noble Bare Steel.

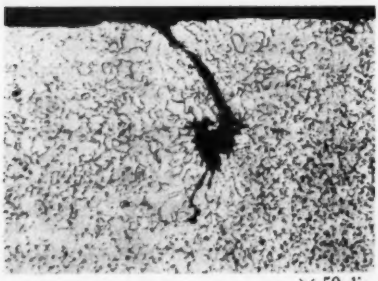

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Fig. 54.—Incipient Corrosion Fatigue Fissures which have Developed during Service from a Fold in a Tube Wall Formed During Tube Manufacture.

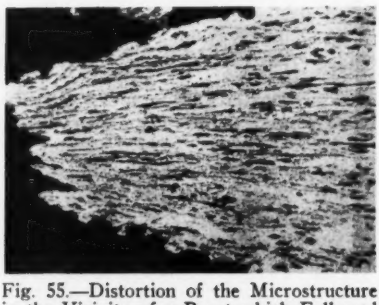

Fig. 55.—Distortion of the Microstructure in the Vicinity of a Burst which Followed Considerable Plastic Deformation by Prolonged Wall Temperatures of approximately 650 deg. C. (1202 deg. F.).

hygroscopic and thus become acid in nature, thereby assisting in the corrosive attack on the adjacent steel.

Corrosion will proceed as long as the boiler is idle and damp, but when in operation, the deposits are dried and attack is temporarily halted. Some oil fuels, depending upon their sources of origin, deposit more sulphurous compounds than others, but they are always present in some measure, and it is for this reason that the systematic removal of accumulated oil fuel deposits should

be carried out at frequent intervals. Boiler design should include provision for ready accessibility of shielded areas, and much is being done in the development of water and steam lances, and soot blowers to facilitate cleaning.

As an immediate means of minimizing this form of deterioration, attempts should be made to prevent the deposits becoming damp when the boiler is shut down, by giving continued attention to rain catchments, leaky tubes, and other sources whereby water gains access into the boiler furnace.

CONCLUSIONS

Deterioration in marine boilers is dependent in appropriate measure on a number of factors involving design, operation and, in particular, the control of water conditions when steaming and standing. Marine operation involves compact and often highly powered units where flexibility and irregular steaming intervals are the rule. Afloat, control of operating conditions and of boiler water lacks the more certain ease associated with that of the shore based power station.

As regards internal (water side) corrosion, the authors are impressed by the fact that the initiation of the damage all too often is consequent on careless attention during idle periods between steaming, and the role of oxygen, gaining access to the boiler water in such periods, must be particularly emphasized. External wastage is also a product of idle periods, and is further accentuated by inadequate soot blowing and cleaning.

Table 1, which classifies and discusses the several forms of corrosion and deterioration, is essentially a summary of the work now under discussion, and taken in conjunction with the several illustrations may be of some practical use to marine boiler designers and users as a ready guide to the subject. It will be noted that many forms of deterioration involve design factors, and the results of metallurgical investigations should provide substantial data from which modifications and improvements may be effected.

ACKNOWLEDGMENTS

The authors carried out the work described above at the Admiralty Central Metallurgical Laboratory, and permission to publish this paper is gratefully acknowledged. Their thanks are also due to Mr. J. W. Messenger, B.Sc. (Eng.), who has assisted in some of the experimental work described and in the preparation of the illustrations.

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MAGNESIUM IN NAVY PLANES

ACKNOWLEDGMENT

This article, reprinted from *Aero Digest* for January 1950, is by CAPT. J. E. SULLIVAN, U.S.N.R., Director Airborne Equipment Division, U.S. Navy Bureau of Aeronautics. It discusses the manner in which magnesium based alloys have demonstrated possible weight saving advantages, increased stiffness and rigidity, and simplification of production design. The article was condensed from an article in the August 1949 issue of *Magnesium*, published by Brooks & Perkins, Inc., Detroit, Mich.

The U.S. Navy Bureau of Aeronautics long has recognized the potentialities of magnesium-base alloys in aeronautical construction, and has been one of the foremost advocates of its wider application in aircraft designs, consistent with the recognized advantages and limitations associated with this important material. From an overall viewpoint, magnesium-base alloys have demonstrated their usefulness in naval aircraft in making possible attractive weight savings, increased stiffness and rigidity, improved safety factor at no increase in weight over aluminum, and simplification of design for production.

The use of magnesium alloys in Naval aviation dates back more than 20 years. By the early '30's, magnesium alloys were already being used in engine castings. These early applications were followed by industrial alloy improvement, Government evaluation, industry-Government conferences on corrosion prevention and programs for development and testing of protective treatments and paint schemes.

North American SNJ-2 *Texan* trainer No. 2008 has been under service test since 1943. when magnesium-alloy outer wing panels were installed. Thirty-two sets of panels have accumulated up to 1800 hours flight time.

Magnesium alloys have come a long way in the past two decades, and their utilization has reached a high degree of success. It appears, however, that a new phase in the application of magnesium is now being entered upon—a phase much broader and more important, in which magnesium will serve as a basis for original and novel design, affording room for the maximum usefulness of this material.

One of the principal problems involved in connection with the extensive use of magnesium in naval aircraft has

been its susceptibility to salt-water corrosion. Accordingly, the earlier applications of this material were concentrated in castings where exposure conditions were such as to minimize corrosion, and where parts were lightly stressed and readily available for inspection.

Since the early days, however, the scope of application has been broadened considerably. Today, the approval of non-critical, non-structural applications of magnesium is routine. Other applications involving primary structure require more detailed study by BuAer from the standpoint of overall suitability, taking into account—in addition to the usual factors of strength, weight, stiffness and fatigue—such considerations as susceptibility to stress-corrosion cracking, susceptibility to corrosion in salt-air environments, toughness and ballistic properties and increased application time and weight of protective coatings.

In designing for the use of magnesium, attention should be given to good stress distribution, elimination of notches and sharp corners, abrupt changes in section, machine marks, and provisions for drainage to eliminate moisture pockets. In the fabrication of magnesium-alloy sheet assemblies, the use of 56S aluminum-alloy rivets is desirable. Spotwelding and fusion welding in inert gases are preferred to other welding methods involving the use of fluxes. In the latter case the flux is required to be in accordance with BuAer specifications.

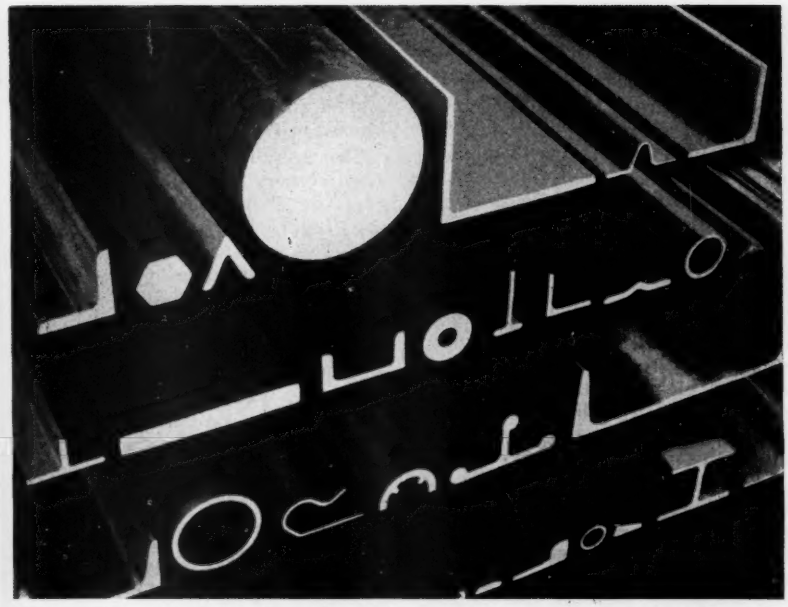
The present use of magnesium in naval planes, based on the above-mentioned considerations as well as on extensive laboratory and controlled service testing, has resulted in attractive weight savings and improved performance characteristics. The applications now generally considered acceptable are as follows: wheels, brakes, fairings, bomb-bay doors, wheel doors, bulkheads and partitions (excluding fire walls), miscellaneous cockpit panels, trailing edges, wing tips, furniture, cabin parts, fittings and similar accessories, miscellaneous housings and covers, spacers, selected ducts, radio and electrical parts, panel stiffeners, air-induction system components, brackets, supports, containers, map cases, a variety of engine parts (both jet and reciprocating), rudder pedals and dive brakes. Cast-magnesium wheels are standard in naval aircraft with the exception of amphibious planes.

In addition to the specific applications mentioned above, the Bureau of Aeronautics has been active in testing entire assemblies made from magnesium alloys. Magnesium-alloy outer wing panels for the North American SNJ trainer have been under service test since 1943. More recently, extensive application of magnesium alloys has been made in experimental aircraft and carried over into the production model. The Chance Vought F7U is an example of this advanced type aircraft using magnesium extensively.

Considerable use of magnesium has been made in highly-stressed applications and, although the detailed construction is of the composite type—employing both aluminum and magnesium in combination in many assemblies—the comparatively large amount of magnesium in the airplane serves as an indication of the possibilities in this direction. Other advance BuAer design considerations contemplate the use of thick skin, monocoque-type construction, which is believed to offer the optimum in structural advantage. This type of construction would find application in wing and fuselage assemblies.

To demonstrate further the possibilities in this direction, construction of a fighter wing assembly has been started. This wing panel is being fabricated for static test purposes and expansion to a number of carrier-based aircraft is contemplated.

Research and development in magnesium alloys and familiarization with the handling and working of this material



Here are shown some of the forms in which magnesium is produced: bar and flat stock, angles, channels, beams, tubing and special extrusions. It is available also in sheet and other forms and for castings.

has been largely responsible, as previously noted, for the expanding utilization of magnesium. Progress would be accelerated significantly by the development of:

1. Substantially stronger alloys, both at normal and elevated temperatures
2. Alloys exhibiting improved corrosion resistance, particularly under stress
3. Alloys with greater toughness, less notch-sensitivity, and less tendency toward crack-propagation
4. Alloys capable of being worked, cast or fabricated more easily.

It is not meant to imply that present materials are deficient in these characteristics to a degree which reduces their usefulness to a low order. To the contrary, the Bureau considers present alloys extremely useful and their deficiencies by no means prohibitive for a large number of important applications as previously listed. However, as advanced application of magnesium alloys develops, the above items assume increased importance. BuAer's research-and-development program, therefore, is aimed at exploring and, if possible, contributing to the improvement of these characteristics in magnesium. This program can be presented best under four headings, namely:

1. New alloy development
2. Studies of existing alloys
3. Structural evaluation
4. Miscellaneous projects.

New-alloy development—The most important alloy development in the program has to do with magnesium-lithium base alloys. These alloys offer several possible advantages over present-day commercial materials, particularly with respect to formability and strength-weight ratio. To date, many hundreds of possible alloy compositions and combinations have been explored. Promising alloys have been based on ***magnesium-lithium-cadmium-zinc-silver*** and ***magnesium-lithium-cadmium-silver-aluminum.*** These compositions and variations thereof have produced alloys ranging from very soft, extremely ductile alloys to relatively high-strength alloys with compressive yield strength in the neighborhood of 55,000 *psi* (as compared with approximately 25,000—30,000 *psi* for present high-strength magnesium alloys). As might be expected in a program of this nature, emphasis frequently has been shifted from one alloy system to another in search for alloys of superior properties. Ductility and temperature-stability very often have been the causes for these shifts in emphasis.

In the early work, the investigation of mechanical properties of these alloys was conducted, for the most part, on extruded bar. Emphasis, more recently, has been placed on the development of hot- and cold-rolling techniques. Severe cold reductions are possible without difficulty, sometimes running as high as 35%. The specific gravity of these alloys is approximately 1.6 as compared with about 1.8 for present-day commercial alloys. Tension and compression moduli average 6.7×10^6 *psi,* and the compressive yield strength generally equals or exceeds the tensile yield strength. Stable alloys have been obtained with ultimate tensile strength in range of 45,000-50,000 *psi,* with the compressive or tensile yield strength averaging a few thousand *psi* lower. The corrosion-resistance of these materials is open to significant improvement and sheet materials therefore may require cladding.

Experimental ingots, billets and slabs have been furnished to several industrial producers for preliminary commercial evaluation. Reports confirm that a very high degree of workability appears to be obtainable.

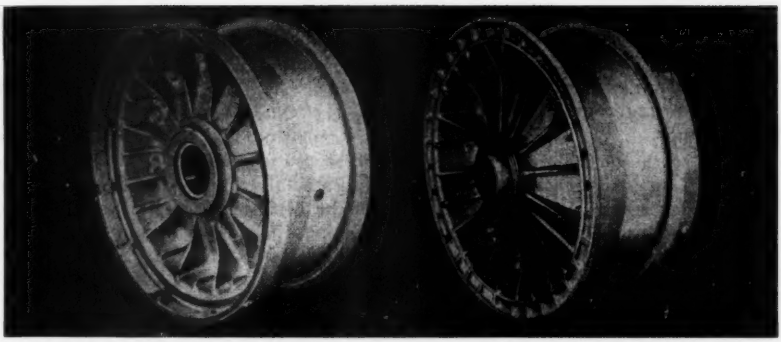
The second alloy-development program concerns ***magnesium-cerium*** base alloys for elevated temperature applications, with significantly reduced creep rate. This work will be continued in the form of evaluation of production-type castings in the improved alloy composition.

Studies of Existing Alloys.—An important phase of the program that has been under way for a number of years concerns corrosion-testing of existing and new alloys. This program is divided into two major categories: one dealing with marine exposure of essentially unstressed magnesium alloys, fabricated and treated in various ways, the second dealing with stress-corrosion phenomena. The former program has brought to light useful information of new combinations of materials and new or modified forms of magnesium alloys.

Very extensive stress-corrosion tests have been conducted on all important alloys to supplement the work done by other agencies. Accelerated corrosion tests (for example, in salt-chromate solutions), plus atmospheric-exposure tests in stress-corrosion racks of various designs, have been used to estimate the maximum stress which these magnesium alloys can be expected to withstand for stated intervals of time under specified corrosion conditions.

For example, it was determined that, under standard conditions with properly stress-relieved material, the AZ31 and AZ51 magnesium alloys, were satisfactory at stress levels approaching the yield strength of the materials.

Another project is under way also to study ballistic properties of magnesium alloys and other aircraft materials. In this project, it is desired to establish standard fundamental test procedures for



Left-and-right view of a typical cast magnesium-alloy naval-aircraft wheel.

evaluating the relative resistance of aircraft materials and structures to damage by gunfire, and to use these procedures in obtaining information not readily available. No data on this project are yet available for publication.

Structural evaluation.—Considerable work has been done at the Naval Air Materiel Center in Philadelphia on the laboratory structural evaluation of sandwich-type construction using magnesium, and on various sheet-stringer combinations. With respect to the former, balsa cores, with facing of several magnesium, aluminum and steel alloys, have been given tests. Although the compressive buckling strength of high-strength aluminum alloys and stainless steel was greater than that obtained for magnesium alloys, sheet-stringer tests, stressing other design factors, showed appreciable advantage for fuselage combinations of magnesium sheet and aluminum stringers.

The service test of 32 sets of magnesium-alloy SNJ wing panels, previously referred to, has furnished valuable information. These airplanes have been in service in coastal atmospheres over a period of approximately five years with as much as 1800 hours flight time. Their panels have received no special maintenance and, although a few difficulties have been encountered, they have given satisfactory service and are in excellent condition.

Recent static tests on panels having well over 1000 hours flying time have shown that wings with AZ31 skin and AZ80 spars and stringers had lost none of their strength. The fatigue strength of AZ51 specimens taken from several panels indicated that the endurance limit had dropped roughly 15%. A project has been established at NAMC to investigate this phenomenon and to ascertain the effect of overstress and understress on the fatigue strength of light alloys.

It is of interest to note that the above airplanes are land-based. Plans are now being made for extensive testing on carrier-based aircraft to determine the effects of differences in handling, exposure, maintenance, and other carrier problems.

Miscellaneous Investigation. A number of investigations continually under way are of interest in the development of the magnesium picture as a whole. For example, a standard welding flux, which has been developed, will enable a better evaluation of other fluxes for gas-welding magnesium.

Work on analytical techniques is also under way for the discovery of such

impurities as iron and nickel which, in even very small amounts, have marked deleterious effects on the corrosion resistance of magnesium. Statistical analysis and spectrographic methods are used in this study.

Other work leading to the establishment of X-ray acceptance standards for magnesium castings has been initiated. It is proposed to define acceptable, borderline and unacceptable type and degrees of defects for areas of high, medium and low stress.

Organic protective coatings for magnesium alloys are being investigated at several Navy laboratories. In addition to exploring the mechanics of making these coating materials, much attention has been given to the selection and study of the vehicles. Phenolics and synthetic coatings have shown excellent characteristics as coatings.

ATOMIC POWER AND FUEL SUPPLY

ACKNOWLEDGMENT

This article, appearing in *Mechanical Engineering* for December 1949, is by MR. WARD F. DAVIDSON, Research Engineer of the Consolidated Edison Co. of New York, Inc. It explores general problems to be considered before atomic power can become a reality, both technically and economically. This paper was presented at the A.I.M.E.-A.S.M.E. Fuels Conference, French Lick Springs Hotel, French Lick, Ind., October 26, 1949.

INTRODUCTION

So much is known about the resources of coal, petroleum, and natural gas in this country that, by comparison, any discussion of the possibilities of utilizing atomic or nuclear fuels is scarcely more than daydreaming. However, the subject is one that deserves attention, if for no other reason than that if even a small part of the wonders that have been forecast for atomic power are realized, the effects on the use of such energy sources as coal may be considerable.

The present purpose is to explore, in a somewhat general way, the problems which must be solved before atomic power can become a reality. There are two aspects to be considered: (1) technical—for until a practical power plant has been built, the rest has no reality; (2) economic—for large-scale use is hardly conceivable unless atomic power plants can produce power at costs competitive with other means.

SOME BASIC FACTS

No method is known, and none has been suggested by competent scientists, for the direct production of electric power from the energy released by nuclear fission. On the other hand, large amounts of heat—equivalent to many thousands of kilowatts—are now being carried off as waste from nuclear reactors used for the production of material for bombs. In one sense the technical problem is that of modifying the reactor so that the heat from it can be used in a turbine to drive an electric generator. This, however, is a great oversimplification. To achieve some understanding of its real character it will be necessary to review certain fundamental considerations.

That certain changes within the nucleus or core of an atom could result in the release of amounts of energy large in comparison with those associated with chemical reactions such as combustion had been known since the discoveries of radioactivity and radium. However, these reactions gave little hope of providing useful sources of power both because the rate of reactions was comparatively slow, and because the rate could not be controlled. In 1939

experiments showed conclusively that some atomic nuclei can split into two fragments of comparable mass with an accompanying release of a large amount of energy which appeared chiefly as heat. In engineering terms, the release was about 4×10^{10} (forty thousand million) Btu per pound, or about three million times as much as from a pound of coal. This gave a new approach. Then, a few years later in a wartime laboratory, it was demonstrated that this "fission" process could be made a self-sustaining or a chain reaction. Until this very important step had been taken the situation had been comparable to that of keeping a fire going with matches, with a gross input of energy greater than the useful output. With a chain reaction, a net useful output became a possibility. That this discovery had not been made earlier is further testimony to the fact that the fission chain reaction can take place only with a very few materials under very special conditions.

The rearrangement of atoms which takes place in a chemical reaction does not change the identity of the atoms. An atom of carbon, for example, remains an atom of carbon whether it is free or combined with hydrogen or oxygen or some other element. The forces that hold the atoms of a chemical substance together are represented as resulting almost entirely from interactions between electrons which move about a central nucleus or sun in much the same way as the planets of the solar system. Most of the mass of the atom is concentrated in this nucleus with considerably less than 1/1000th part in the electrons.

In fission, on the other hand, the nucleus of the atom splits into two parts forming nuclei for two new atoms. If the original atom is U-235—that is, the isotope, or species, of uranium atom with a mass of 235 units—the fragments might be krypton and barium, or some other pair with about the same total atomic weight. This is a fundamental difference from a chemical reaction, and the one which explains the proportionally greater amount of energy that is released.

Greater importance, for the immediate discussion, attaches to another difference. Fission is initiated by a neutron—an electrically uncharged or neutral particle—entering the nucleus of a uranium atom. The neutron is the match or detonator that sets off the fission. On fission the two parts of the original nucleus fly apart with tremendous velocities, and from one to three neutrons are ejected. It is these neutrons with which we are most concerned, for if conditions are right they may enter another uranium nucleus and so set off a new fission reaction. If, on the average, exactly one neutron from each fission causes a new fission, the rate will remain constant; if the average is larger or smaller than one, the rate will increase or decrease with corresponding rapidity. At one limit the action becomes explosive, and the rate of energy release is so great that the temperature may be expressed in millions of degrees and the time in microseconds (millionths of a second); at the other limit the reaction ceases quite abruptly. The chief engineering interest lies of course in the midrange or near the stable or constant-rate condition. Reduction in the rate may always be achieved by inserting into the zone where the reaction takes place some nonfissionable material which will absorb neutrons and thus reduce the number which remain to produce further fission; increase in the rate of reaction may be harder to achieve.

FUNDAMENTAL DESIGN PROBLEMS

Some of the most difficult problems of nuclear-reactor design arise in the imperative need to conserve neutrons so that an adequate supply will be available to maintain the chain reaction, with a small excess to permit control. There are several ways in which neutrons can be lost:

1. By escape from the reactor, for even the best neutron reflectors are only moderately effective.
2. By absorption by "impurities" in the materials of the reactor.
3. By absorption by U-238 without causing transmutation into neptunium.
4. By absorption by fission products.
5. By absorption by structural materials and heat-transfer fluids within the reactor.

Loss of neutrons by escape is less, percentagewise, for large-reactors than for small ones, and is less for spherical reactors than for square or elongated ones. Also, it is less for compact or high-density reactors than for low-density reactors where a smaller proportion of the space is filled with fissionable material.

Absorption of neutrons by impurities in the materials of the reactor may be large and sometimes necessitates taking very special precautions in the selection of materials.

Absorption of neutrons by U-238 is partially compensated for in some reactors by the transmutation which leads to the formation of Pu-239. This will be discussed later.

To limit the absorption of neutrons by fission products, it is necessary to remove the fission products from time to time. As these will exist as an intimate mixture with the uranium, their removal requires taking the whole mixture—uranium and all—out of the reactor and subjecting it to complex chemical treatments to separate the waste from the unspent uranium which, for large-scale power, will be too valuable to throw away. Since the whole mixture is very radioactive, the processing plant must be designed for almost wholly automatic operation and to permit remote control of those operations which are not automatic. The chemical plant may be remote from the reactor, but the design of the reactor must provide suitable means for changing the charge of the active materials.

The need to limit neutron losses through absorption by the structural materials and the heat-transfer fluids places two heavy restrictions on the designer. For one, almost none of the common engineering materials is acceptable because of the high specific absorption or large "capture-cross section," and alternatives must be sought. For the other, the quantities of materials must be limited severely, and this means using them at high stress.

THE PROBLEM OF MATERIALS

The search for materials which meet these requirements is, in itself, a task of major magnitude. Even when the physicist has furnished data on the capture cross sections of particular elements, only a start has been made. Few materials are suitable, in their pure state, for engineering application; most useful materials are alloys, just as steel is an alloy of iron and carbon with a number of other elements. Equally, small amounts of impurity may make the material useless for engineering purposes and, as we shall see, impurities may sometimes be formed within an alloy as a result of exposure to radiation. The investigation of these factors is both costly and time-consuming. After the metallurgical laboratories have developed some alloys which satisfy the

first set of requirements, it becomes necessary to investigate suitable methods for production and fabrication on an engineering scale. Aluminum was hardly more than a laboratory curiosity until the Hall process was invented, and commercial production became possible.

After a new material has been produced in trial processes the engineer must examine it to determine how it can be used. Just as a part to be made of cast iron must be shaped differently from one to be made from a chrome-nickel alloy steel forging, so one to be made from one of the new alloys—as yet unformulated—will require its particular shape and proportions, especially if the amount of material is to be kept at a minimum. Then the design and material, and the fabrication processes, must be verified to insure adequate resistance to corrosion and to fatigue, for an unusually high degree of reliability is essential in reactor components.

This brings up another group of problems. It is known in a general way that every material undergoes some change in its physical properties when subjected to neutron bombardment and radiation of the intensities existing in a nuclear reactor. However, only very limited data have been obtained, because adequate facilities for making the studies have not been in existence. Moreover the subject is a very complex one because in this case, again, it is not enough to study the behavior of pure metals; the alloys themselves must be studied. It was noted before that most engineering materials are alloys, and it is emphasized that the valuable properties of alloys often depend on the amount and disposition of one of the minor alloying elements. For instance, the total of all the alloying elements in high-strength aluminum rarely exceeds 6 or 7 per cent, and yet the ultimate tensile strength of the alloy may be 2 or more times that of commercially pure aluminum. The tensile strength and resistance to fatigue and corrosion of such an alloy might be impaired seriously by the alteration in the silicon content due to radiation, even though 90 per cent or more of the aluminum remained unaffected.

The problems to be assigned to the metallurgists are, therefore, both numerous and formidable. Before we leave the subject of materials and turn to problems more in the field of engineering, we should not overlook the possibilities of help from the ceramics experts. Ceramics and glass, although among the oldest materials made by man, have been little used for engineering purposes until recently, and even now their potential uses are only partially understood. However, ceramics cannot be looked on as direct substitutes for metals. To realize their full potentialities, the engineer must be ready to make quite radical changes in his design.

NUCLEAR FUELS

Before proceeding to a consideration of some of the specific engineering problems of reactor design, we must divert to a short discussion of fissionable materials and nuclear fuels.

The technical problems of both the designer and the operator of fuel-burning boilers would be made much simpler if basic fuels such as coal were first converted into gas in a separate plant, about which the power man had no concern or worry. Of course in a practical way, such a solution of troublesome problems is never possible, for the question of costs—the total costs—cannot be ignored. In much the same way power-reactor design could be made far easier if the designer could contemplate the use of concentrated uranium-235 or plutonium-239 rather than natural uranium, which contains only 0.7 per cent of fissionable U-235 and

about 99.3 per cent of nonfissionable U-238. Such a solution again evades the question of cost, and it also evades in some degree the question of adequate supply.

The early designs for atomic bombs contemplated the concentration of the U-235 in natural uranium, and several elaborate and costly "isotope separation" plants were built, mostly at Oak Ridge, for accomplishing this.

Experience with these processes has shown that the power requirements are so large that the amount of energy used in them is comparable to that available for power purposes in the finished product. Large-scale power production—which is our main interest here—if based on such methods would merely introduce a new step in the cycle and not result in any reduction in the use of coal, oil, natural gas, or water power. Clearly, such arguments as this are of minor importance for a limited number of special power applications such, as for the propulsion of naval vessels, and they are without force when considering bomb materials.

It is fortunate that the possibilities do not end here, and that nuclear reactions are known which give promise of making it possible to utilize the U-238 as well as the U-235. These reactions, as used in the processes for the production of bomb material, involve several steps. In the first step, a neutron released by the fission of an atom of U-235, after being slowed down by successive collisions with nuclear particles such as the nuclei of carbon atoms, enters an atom of U-238, which then becomes an atom of U-239. In the next step the U-239 emits a beta particle, or electron, to become an atom of neptunium-239, and then, by another like radioactive transformation, the Np-239 transmutes into plutonium-239. This end product is fissionable in the same way as U-235, and it can be used to "recycle" the process. Moreover, since more than one neutron, on the average, is released on the fission of each atom of U-235, the possibility exists, if the loss of the neutrons can be made small enough, that the number of new atoms of Pu-239 will be greater than the number of atoms of U-235 destroyed by fission, and that there may be an actual increase in the quantity of fissionable material, with the nonfissionable U-238 providing the substance or mass. There is also an analogous series of nuclear reactions by which fissionable U-233 can be produced from nonfissionable thorium-232. These "breeder" reactions offer the hope of being able to increase the potential supplies of nuclear fuels by a factor of 140 or even more.

However, as so often happens, this gain is not without its price. To get the maximum breeding of new fissionable materials, neutron losses must be kept to the very minimum, and this is incompatible with meeting the design requirements for a power reactor, particularly as to those for heat transfer and for control.

GENERAL REACTOR DESIGN

With a tentative list at hand of materials which do not have large neutron capture cross sections and which show adequate stability when exposed to radiation, and with some decision as to the kind of nuclear fuel to be used, the engineer may make a start by outlining a preliminary design.

Extreme compactness is essential to insure needed conservation of neutrons, and this means high rates of heat transfer and high mechanical stresses, both of which are incompatible with a high degree of reliability. Because of the extreme difficulty, which may amount in some cases to the absolute impossibility, of making repairs to a reactor after it has been put in operation and has become radioactive, every practical means must be taken to insure

against failure of any parts. But as perfection in this is unattainable, the design must make provision for some maintenance and repair work.

At this point we will return, for purposes of illustration, to the Hanford type of reactor, keeping in mind all the while that the final design may be for a quite different reactor.

In the Hanford reactors, uranium bars, sealed in closely fitting aluminum tubes, are inserted into other tubes which line holes running through blocks of graphite "moderator." Water is circulated through the annular space around the "slugs" and inside the tubes to carry off the heat. Although the total heat is equivalent to many thousands of kilowatts it is not usable in a turbine because the temperature is too low.

A basic concept of great importance in designing heat engines—and steam or gas turbines are heat engines—is known as Carnot's principle. This states that the maximum efficiency at which any simple heat engine can transform heat into mechanical power is dependent on the range of temperature of the working fluid (steam or gas) according to the formula.

$$\eta = 1 - (T_2/T_1)$$

where T_1 is the initial (or higher) temperature and T_2 is the final (or lower) temperature. For this use the temperatures must be expressed on the absolute scale for which the numbers are 460 greater than for the Fahrenheit scale.

To see more clearly what this means, we may assume quite favorable conditions, as with cooling water from a river or lake available for condensers at 75 F (535 R). With steam at 350 F (corresponding to a pressure of 120 psig) the maximum attainable efficiency would be 25 per cent, and the actual efficiency considerably below this figure because of unavoidable mechanical and heat losses. With air cooling, as would be used with many gas turbines, the final temperature might be above 300 F, and to attain an actual (rather than the theoretical) efficiency of 25 per cent the initial temperature would have to be above 1000 F.

One way to get higher temperatures from the Hanford reactors would be to reduce the flow of water until there was steam generation. This, however, is open to a number of very serious objections, prominent among which is the fact that the steam, because its density is lower than that of water, will have a lower specific absorption for neutrons and this difference may be great enough to cause very serious disturbances in the operation of the reactor. Increasing the pressure so as to permit reaching higher temperatures without steam formation is not an immediately acceptable alternative because this would necessitate thicker walls for the tubing and a greater weight of inactive metal in the reactor.

HEAT TRANSFER A PROBLEM

An evaluation of possible heat-transfer fluids presents complex and difficult problems because of the many factors that are involved. Gilliland [1] has made an interesting start. Taking into account various physical properties such as specific heat, thermal conductivity, density, viscosity, which influence heat transfer and the power necessary to circulate the fluid, he has developed two sets of parameters which give measures of the relative merits of different fluids. He then calculated the values of these parameters for 21 gases and vapors and for 15 liquids and molten metals. A study of these values and of data on neutron capture cross section led him to select four media for further study. They are helium, water (in spite of the objections noted), molten cadmium, and

molten lead. Other investigators have added air and some eutectic mixtures involving lead.

The inclusion of molten metals in the list may seem strange because it is obvious that they cannot be used directly in a turbine or heat engine. The justification comes from the fact that a "two-stage" heat-transfer system probably will be necessary in any case. With this system one fluid, which does not vaporize, circulates through the reactor where it picks up heat and then through a steam generator or heat exchanger where it gives up that heat to make steam which will be used in a steam turbine or to heat gas which will be used in a gas turbine. This arrangement has the great and overriding advantage of providing isolation of the turbine and its auxiliaries from the radioactivity which the fluid circulating through the reactor will acquire. Without it, turbine parts would soon become radioactive or so contaminated by radioactive materials that usual procedures for making operating adjustments and repairs would have to be replaced by others using remote-control robot devices. It does not, however, wholly avoid problems of this kind for there must be pumps or blowers for circulating the fluid in the first circuit and these must tolerate the radioactivity.

Before a final selection of the heat-transfer fluid can be made it will be necessary to re-examine the question of structural materials and particularly those used in the active zone of the reactor. In addition to the requirements previously noted, the materials must meet some others. They must not be corroded or attacked severely by the heat-transfer fluid; they must have low specific neutron absorption; they must have adequate strength, especially at high temperatures; they must have adequate thermal conductivity; they must be capable of fabrication; they must have good stability when exposed to radiation. No common engineering material begins to meet all these requirements, which are sometimes conflicting, and progress in the development of power reactors is dependent in a large degree on the success of the metallurgists in meeting the challenge thus put to them.

MEANS FOR REPLENISHING THE CHARGE

Another detail of reactor design which requires careful study is the means for removing the partially spent charge and replacing it with new active materials. The fission products are generally active neutron absorbers so it is important that they not be allowed to accumulate unduly. Also depletion of the charge reduces the level of activity so that the chain reaction will cease before all the charge has been used up. In the Hanford reactors, bars or slugs of uranium, in their aluminum cases, are removed from time to time and replaced by fresh ones. Such a process seems, however, to be ill-adapted to a power reactor, and a continuous process of some kind would offer many advantages if it could be devised. Such a process might be deemed as analogous to the continuous feed of coal in a stoker or in a pulverized-coal burner which has replaced intermittent hand firing in modern coal-burning power plants. The partially spent charge would be sent to a chemical processing and reclaiming plant, as was noted before.

CONTROL

Basically, control of the reactor is accomplished by moving rods of neutron-absorbing material into or out of the active zone. Cadmium has been used generally because it is a particularly effective absorber of neutrons, and therefore the control rods can be small. Actually, the control presents many very difficult problems. The control of the neutrons so as to control the fission rate cannot be concentrated at one convenient point, but must be distributed throughout the reactor if local zones of excessive temperature are to be avoided. The control system also must be responsive to changes in neutron flux density which take place rapidly, and not depend wholly on temperature of the heat-transfer fluid, or on changes in steam pressure which take place much more slowly. Additionally, the control system must include a large number of instruments and detectors to give warning of failures of one kind or another, such as the development of leaks of heat-transfer fluid, or the escape of radioactive gas.

Therefore the control and protective systems will require a large array of instruments and servomechanisms. Some will be of rather conventional type, possibly modified to give better life when exposed to radiation, but a large part will require new developments. That the problems can be solved is demonstrated by the successful operation of a number of large reactors used for the production of bomb materials, yet the opinion has been expressed in a recent paper [2] that finding a satisfactory solution may be crucial to the future of nuclear power.

SHIELDING

The need for heavy shielding to protect personnel and equipment from injury or damage due to radiation has been mentioned in nearly every discussion of atomic energy. Gamma rays, which are similar to x-rays, and neutrons are the most difficult to deal with for they are not stopped completely by any thickness of shield, and the design must be based on a tolerable residual intensity. In designing the shielding for a reactor, a large amount of ingenuity is necessary to devise means for insuring the effectiveness of the shield and, at the same time, providing means for making connections, installing instruments, and so forth. Provision must also be made for essential maintenance operations.

The shielding problem, however, extends beyond the reactor. The heat-transfer fluid will become radioactive and, in turn, may induce radioactivity of somewhat less penetrating types in materials with which it comes in contact. Thus the first heat-transfer circuit and all the equipment associated with it will require some shielding, again with provision for access for making maintenance repairs.

So long as the heat exchanger or steam generator does not develop any leaks, the tube walls will form an adequate barrier to isolate the steam or gas system, but it will be unsafe not to make some provision for dealing with the situation if a leak should occur.

STEAM GENERATOR AND SOME AUXILIARIES

The steam generator itself presents some new problems. For one, the need for keeping to a minimum the volume of fluid in the first heat-transfer circuit points to the desirability of using high rates of heat transfer, and this, in turn means that particular care must be taken to insure continued cleanliness of the

surface and freedom from sludge and scale deposits. For another, it means that there must be carefully planned arrangements for treating the feedwater and conditioning the boiler water. It means, too, provision of adequate facilities for inspection and periodic cleaning.

If molten metal is used as the heat-transfer fluid, provision must be made for draining the system on shutdown or for melting the metal in place when starting up. If air is used, attention must be given to the safe discharge back into the atmosphere, for there will be a moderate amount of radioactivity. Extensive studies dealing with this last problem are now under way at the Brookhaven National Laboratories.[3]

We now turn our attention to the power plant. What will it look like? There are no compelling reasons why the turbo-generators and the switchgear should differ greatly from those of fuel-burning power stations. Probably they will be a little larger because of the advantages of large reactors, but that is about all. The real differences will be in the steam-generating equipment and the facilities for fuel-handling.

Instead of large steam generators, often of impressive height, there will be a much smaller cubic reactor encased in a massive concrete and metal shield and another unit, the steam generator in a less massive shield. There will be circulating pumps to handle the molten metal heat-transfer fluid or blowers if air is used. In either case these will be large units, for present studies all indicate that the power requirements will be large because of the high velocity at which the fluids must be circulated through constricted passages. Boiler feed pumps will be conventional.

There will be no coal-handling equipment, no coal pulverizer, no air preheaters; there will be no dust collectors; there will be no ash-sluicing or slag-tap system. Instead there will be massive containers, probably lead-lined, for handling the partially spent charge and removing it to the chemical plant; there will be less massive containers for handling the replenishing charge.

The control room will be more elaborate than in the fuel-burning counterpart, and will have more interlocks and automatic features.

COSTS

What will this power plant cost? A large high-efficiency coal-burning power plant today will cost something like $190 per Kw. Of this, about one-half or, say, $100 per Kw would be needed for the part that would be duplicated in a nuclear power plant. There is not an adequate basis for a sound estimate of the other costs, but it may not be unreasonable to assume that they will be at least 3 times those of the coal-burning station. This gives a figure of $370 per Kw for the complete nuclear power station.

How will operating costs compare? The largest item for coal-burning stations is fuel. For the nuclear station the costs could conceivably be far lower, if—and this is an important "if"—ample supplies of uranium ores are found and if it is found possible to reduce greatly present costs of reclaiming and reprocessing the partially spent uranium. (At some later date, thorium may enter the situation as an alternative to uranium.) About the best that one can do now is to say that under present conditions uranium as a fuel for power plants is much more costly than coal, but at some later date it may become somewhat less expensive. As to other operating costs, it seems clear that until there has been extensive experience with nuclear power plans, labor supervision, maintenance, and repair are all likely to be higher than for coal or oil-burning stations.

SOME FACTORS INFLUENCING COSTS

It must be kept in mind in any attempt to draw conclusions from these figures—and they are at best only the roughest sort of guess—that they have been set up on the assumption that the power plant was single-purpose development. If, as seems not unlikely, power plants are built as part of a multipurpose development, then the allocation of common costs, which is to some extent quite arbitrary, may alter the picture.

There are, however, some additional considerations. In making a comparison of nuclear and fuel-burning power plants, it should not be assumed that they can be placed on sites equally advantageous with relation to the load. Until years of successful operation have proved their reliability, no prudent engineer would place a large nuclear power station in the heart of a large city or in densely populated or industrially very important areas. The ever-present chance, however small, that something might go wrong and release large amounts of intensely radioactive material would dictate sites in areas where a failure would be less disastrous. This may necessitate added transmission, the costs of which must be considered.

The large size of nuclear power plants may operate to their disadvantage in regions of low load density, for no single small area such as a city may have enough load to justify such a station, and adequate load could be found only by supplying other areas through a transmission system, the costs of which must be taken into account.

The disposal of radioactive wastes presents many problems for which solutions may be found only by distance from centers of population. This does not mean that the power plants must be located in remote regions, for there is the possibility of shipment in suitably shielded cars or ships, but the costs of such operation cannot be ignored.

Almost regardless of what costs may be for nuclear fuels, the energy per unit weight is so great—about 3,000,000 times that of coal—that shipping costs of the uranium will be practically negligible as compared with those for coal or petroleum or transmission costs for natural gas. (This does not apply to the partially spent charge which must be handled in very heavy shields.) Therefore, in the absence of tariff barriers and artificial price control, the fuel cost will be the same at any place in the world. Aircraft and surface vessels might, in theory, be made free from the high price they must now pay for carrying their fuel with them.

This advantage must not be confused with one that has sometimes been claimed for nuclear power plants. It has been suggested that they offer particular advantages in areas where adequate supplies of condenser cooling water are not available in lakes or rivers, but this does not meet the test of critical analysis. A heat engine (steam turbine or gas turbine) is required in either a nuclear or a fuel-burning plant and the conversion efficiency will be influenced equally by the availability or lack of cooling water. Higher final temperatures mean lower efficiency, larger turbines, larger steam generators, larger reactors, and so forth, and they mean higher fuel consumption. All that can be said with certainty is that at some time in the future nuclear power plants may not be influenced as adversely as fuel-burning power plants by lack of natural cooling-water supplies.

SUMMARY

There appear to be no unsurmountable technical barriers to the development of large nuclear power plants, but it is by no means clear that such plants can offer anything like as great economic advantages as have often been predicted. In fact, the outlook as to costs, even if the enormous development costs were to be charged entirely to other projects, is such that one would not expect rapid large-scale inroads of nuclear power into the field of industrial power production. At most, nuclear power plants might represent most of the new capacity and thus be a supplement to rather than substitute for power from coal, oil, gas, and water power.

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Official U. S. Navy Photograph

Maneuvers of Motor Torpedo Boats—U.S.S. PT.-105.

AN ATTEMPT TO FIND AN EXPLANATION AND WORKING THEORY OF CYLINDER LINER WEAR, COMPARED WITH PRACTICAL OBSERVATIONS

ACKNOWLEDGMENT

MR. H. BERGENHEIM, Chief Engineer of the Broström Lines, Gottenburg, is the author of this article which appeared in the December 1949 issue of the British Motor Ship.

This paper contains the conclusions I have reached in trying to find an explanation of the causes of the abnormal cylinder wear in ships' Diesel engines, mainly of the two-stroke type. It is thus not a question of dealing with the causes of what is usually called ordinary cylinder wear, by which I mean wear on the cylinder liners which keeps below 0.1 mm. per 1000 running hours.

I shall first review the three different processes which can be assumed to take place during the combustion in an internal-combustion engine cylinder, and the consequences I believe these processes generally have with respect to the condition of the combustion spaces. These three processes are the following, viz:—

1. Electrochemical.
2. Oxidation.
3. Detonation.

On the bases of these assumptions I present:

4. An attempt to establish a working hypothesis, and then compare this proposition with the following tests performed by our company, viz:
5. Corrosion experiments.
6. Voltage measurements in experiments with an acetylene flame as source of heat, and with our investigations and measurements in the ships, viz:
7. The working hypothesis compared with observations on board ship:
   (a) The cylinder wear in relation to the insulation resistance in the electric installations on board.
   (b) Measurements on board the ships.
   (c) Four- and two-stroke main engines and auxiliary engines.

I shall next give an account of

8. Examination of the Diesel oil, and finally
9. A summary of my findings.

1. ELECTROCHEMICAL PROCESS IN AN OIL-ENGINE CYLINDER

The humidity of the air in combination with the sulphur content of the fuel oil forms steam and sulphurous acid in the engine cylinder during combustion. When the piston during the working stroke moves downward, *and* these vapors come in contact with the comparatively cold cylinder surfaces, the vapors wholly or partly condense and mix and form electrolyte. If the relative humidity of the air is 90 per cent at 20 degrees C., and the oil is burned with 75 per cent excess of air, the hydrogen content of the oil is 12 per cent and the sulphur content 1 per cent, there will be developed (per ton of oil consumed) about 1500 kilograms of steam and 25 kilograms of sulphurous acid.

If a metal comes in contact with water or a solution it will send out positively charged ions from the metal surface into the liquid. These ions are formed because the metal atom, upon separating from the electrode, leaves one or more electrons on the electrode. Through an excess of electrons the metal will be negatively charged, and this state is increased with the number of ions sent out into the liquid. If electrons are not conducted away from the metal, a state of equilibrium will soon be established, after which the liquid cannot receive any more ions. A fixed difference of potential (the electrode potential) is then attained between the metal and the liquid, i.e., the voltage existing between the metal and the liquid when the solution will not absorb any more metal ions. The same condition may be conceived as existing between the ship's hull and the water, and in any engine cylinder between the cylinder liner and the electrolyte.

Now, if the metal is subjected to electric tension, the electrons will be conducted away gradually, and we do not attain the electrode potential. Instead there is a continuous release of electrons.

Should my assumption prove correct, the consequence will be that the electrons which should have been stored in the engine material participate in the movements of the electrons that take place in the electric plant. In these circumstances all metal in contact with water or an electrolyte continuously flings out metal ions into the liquid, the visible result of which we call corrosion of the metal.

The process described must be regarded as the main element in the processes then started in the engine cylinder according to the assumption made, and which in what follows I shall call the oxidation process and the detonation process.

2. OXIDATION PROCESS IN AN ENGINE CYLINDER

H. Schönfeld and others have pointed out that hydrocarbons can be oxidized through the action of an electric current in the presence of sulphuric acid. In the course of this action are precipitated larger or smaller quantities of sebacic acids, depending on the existing prerequisites.

It is further ascertained that a hydrocarbon under conditions favorable for oxidation forms alcohols, ketones (associations of the radical CO with two alcohol radicals of type R CO R^1), aldehydes (acidulous alcohol products), and superoxides.

According to the assumption of the process in an engine cylinder in case of low insulation resistance of the electric plant in the ship, the cylinder oil will be exposed in the cylinder to the oxidizing influence of the electric current. Oxidation products will then be produced out of the hydrocarbons in the cylinder oil, which subsequently owing to the pressure and temperature condi-

tions in the cylinder may be further oxidized to form pitch and resin-like substances. In turn these substances cause the cylinders and pistons to be prematurely soiled and coated. The oxidation of the cylinder oil moreover reduces its lubricating properties and thus indirectly contributes to the increased cylinder wear.

Some of the oxidation products have also an unfavorable effect on the combustion process in the cylinder. I shall revert to this in my review of the detonation process.

3. DETONATING PROCESS IN AN ENGINE CYLINDER

In detonation tests performed in steel tubes of from 7 to 13 mm. diameter and 1.6 to 6 meters' length, the detonation wave attained a speed of up to 2100 meters per sec., the return wave up to 1700 meters per sec., and the detonation wave a temperature of 5570 degrees C. (Cf. A. Egerton and S. F. Gates, The Royal Society, London, Reports, series A, vol. 114, No. A766, and Harold A. Beatty and Graham Edgar, "The Science of Petroleum.") The detonation phenomenon is not fully explained, and I must here confine myself to dealing with detonations as a reality and try to estimate the consequences of this phenomenon occurring in an engine cylinder.

For the process in an engine cylinder which I have in mind, the expression "detonation" is perhaps misleading, inasmuch as a fully developed detonation can hardly take place in a space with so large a diameter in relation to its length as a Diesel-engine cylinder. It will in any case represent a detonation of a lesser order.

Literature on the subject mentions that detonating processes can occur in an engine cylinder if substances such as the oxidation products mentioned in the foregoing, i.e., aldehydes, superoxides, etc., are present during the combustion. As an illustration it may be mentioned that one molecule of such a substance can affect 200,000 molecules of the fuel.

Scientists state inter alia that with combustion of a detonating character, higher temperature, greater rapidity of the combustion wave and more ultra-violet light are produced than at normal combustion.

One may presume that the condition will be the same with combustion of a detonating character, as compared with normal combustion in an engine cylinder, even though temperature and velocity do not attain such high values as in a fully developed detonation.

The uncontrolled temperatures obtained nevertheless are apparently high enough to damage the surface layer in the cylinders. The uncontrolled combustion takes place in the same part of the cylinder as the normal combustion, namely in the upper part of the cylinder where there is the maximum wear.

In trying to think of some phenomenon which would afford theoretical support of the assumption of detonating occurrences in a ship's engine cylinder I have proceeded from a case well known in radio technics, namely that a metal plate which is exposed to ultra-violet rays, or is heated, is able to send out electrons. As both ultra-violet light and high temperature would be produced in a detonating process in a cylinder, it is conceivable that the result would be a release of electrons from the cylinder surface, and that electric currents are obtained in the combustion space. The voltage of these currents would then be a criterion of the violence of the detonating process.

4. ATTEMPT TO FORM A WORKING HYPOTHESIS

On the basis of the foregoing observations a tentative working hypothesis concerning the combustion process in the cylinders will now be presented.

By reason of the presence of electrolyte in the cylinders, metal ions are being flung out from the cylinder liner surface. Electrons are released.

If the electric insulation in the ship has low resistance the released electrons are conducted away and released metal ions are continuously flung into the electrolyte. The visible result is corrosion of the metal.

The electrons which are released and conducted away from the cylinder surface, furthermore produce oxidation of the cylinder oil. Oxidation products are formed which for one thing coat the cylinder surfaces and for another start combustion processes of a detonating character.

This detonating process gives rise to formation of an electronic current through the engine and propeller shaft to the ship's hull and the sea.

5. CORROSION EXPERIMENTS

In order to ascertain to what extent D.C. tension can affect a corrosion process, we have performed a few simple tests.

In each of six equally large and well-insulated glass vessels was poured electrolyte composed of 1 dl. sulphuric acid and 2 dl. water. Six copper bars were weighed separately and immersed simultaneously each in one of the vessels. The bars were connected as follow:—

1. Unconnected.
2. Connected to earth.
3. Connected to the pos. pole of a 220-v. D.C. generator.
4. Connected to the neg. pole of a 220-v. D.C. generator.
5. Connected to the pos. pole of a 220-v. D.C. generator.
6. Connected to the neg. pole of a 220-v. D.C. generator.

The electrolyte was changed three times and the bar wiped clean six times.

The test was concluded after 96 hours. The bars were then weighed and the loss of weight noted.

It was found that the two bars connected to the pos. pole and neg. pole respectively, and left undisturbed during the test, had lost about 20 per cent, more weight than the unconnected bar and the bar connected to earth. In the cases of the bars where the electrolyte was changed and the protective coating on the metal wiped off—intended to illustrate the process in an engine cylinder—the result was a reduction of weight about 100 per cent greater.

The test proves that the corrosion of the metal is very much increased in the presence of electric D.C. tension. When the electrolyte was changed and the metal wiped clean the corrosion was increased many times.

6. VOLTAGE MEASUREMENTS IN EXPERIMENT WITH ACETYLENE GAS FLAME AS A SOURCE OF HEAT

Experiments have further been carried out with the object of trying to demonstrate the processes during combustion of a detonating character in an engine cylinder, as well as to give an idea of the electric tensions set up in this connection.

In view of the occurrence of higher temperature and more ultra-violet light during combustion of this nature than

in normal combustion, an acetylene-gas flame was used in these experiments as source of heat. In such a flame there is present both high temperature and ultra-violet light.

The experiments demonstrated that if the flame was turned against steel or cast iron, currents were immediately produced, which were visible in a cathode-ray oscillograph when connected with the test piece, and could be measured with a millivoltmeter.

The negative terminal of the voltmeter was connected with the test piece, and the positive terminal with the metal in the gas burner. Voltages of from 100 to 400 mv. were registered. The current immediately disappeared when the flame was removed from the test piece.

With the cathode-ray oscillograph, voltages could be observed when the gas flame was rapidly passed back and forth over the test piece. Doubts were entertained about the reactions of the oscillograph, since its function is to measure A.C. tensions and frequencies. The fact that, as mentioned, it did react must, no doubt, be attributed to the rapid alterations of temperature in the gas flame, accompanied by release of electrons at the same rate. From this is obtained a pulsating continuous current, which is able to act on the condensers of the oscillograph and make it register a voltage which has the appearance of alternating current.

EXPERIMENTAL ENGINE

In order to enable some preparatory demonstrations and tests to be performed before commencing measurements in ships, a two-stroke Diesel engine was procured and erected for certain planned experiments ashore.

7. WORKING HYPOTHESIS COMPARED WITH OBSERVATIONS MADE ON BOARD DURING THE RUNNING

It may naturally be asked at this stage how far the tentative hypothesis agrees with observations in the actual operation of Broström Lines' ships. Based upon the investigations made, an account will now be given of our experiences.

APPENDIX

Fig. 1 is a graph showing the average cylinder wear in the main engines of ships belonging to the Broström Lines during the years 1935-1948.

The average cylinder wear was already high in 1935, and there was a considerable reduction in the years up to 1941, in which latter year a minimum was recorded.

From 1942 the curve rises slowly at first and then with a steepness never before recorded. This rapid increase was arrested in 1946. Since then the cylinder wear has certainly increased, but not at the previous extraordinary rate.

Thus, in 1942, the average cylinder wear was 0.112 mm. per 1000 running hours. By 1945 it had risen to 0.239 mm., and, finally, in 1948 to 0.314 mm., an increase since 1942 of about 180 per cent.

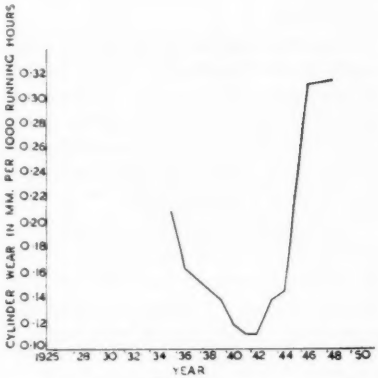

Fig. 1.

7A. CYLINDER WEAR IN RELATION TO THE INSULATION RESISTANCE IN THE SHIP'S ELECTRIC SYSTEMS

In the course of our examination of the quality of insulation in the different ships, we have found that in ships in which the insulation is poor the cylinder wear has been great, and that ships in which the insulation resistance is higher have smaller cylinder wear.

It is often noted that on the inspection of pistons and cylinders these are considerably more dirty in some engines than in others even though they are of the same type. With closer control it was found that in the ships where the cylinders were most dirty the insulation of the electric plant on board was rather weak. As a rule the fouling was less conspicuous in ships with higher insulation resistance.

Both of these observations undoubtedly furnish a certain support to the assumption of a correlation between the quality of the insulation and the cylinder wear. In general it is the positive pole that has the lowest insulation resistance.

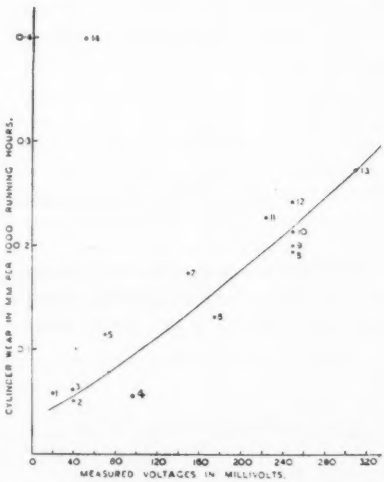

Fig. 2.

7B. MEASURINGS ON BOARD

Three different instruments have been used for determinations of the detonation process just mentioned, viz.,

Cathode-ray oscillograph for measuring frequencies and any stored pulsating D.C. tensions that may rise, and

Millivolt meter (with a sensitivity of 20,000 ohms per volt) and Tubular voltmeter } For measuring D.C. tensions

(Instruments of this degree of sensitivity are required for such records.)

The oscillograph registered voltages like the voltmeter but as a rule gave higher values. The frequency values obtained with the aid of the oscillograph, however were difficult to read with any degree of safety, and have consequently not been included in any record. Thanks to the oscillograph we have nevertheless been able to ascertain that the combustion process is very different in different engines, and that each individual combustion gives rise to distinct and marked tension developments.

The difficulty connected with carrying out measurements in the combustion space of the cylinders has induced us to look for more convenient measuring places. It then occurred to us that owing to the insulating properties of the lubricating oil the cylinders and frame of the engine as well as the hull are insulated from the pistons, connecting rods, crankshaft and tailshaft. By this reasoning we would be justified in connecting our voltmeter between points situated in those parts insulated from each other, and for the sake of accessibility we selected the propeller shaft and the ship's hull.

Only the values indicated by the millivolt meter which are compiled in graph, Fig. 2, are used together with the recorded cylinder wear figures for the ships in question.

It will be seen from the graph that the measured tensions in the main agree with the amount of cylinder wear in the ships. The voltage of the currents, in other words, may be a criterion of the violence of the detonation process.

An observation of interest in this connection is that the voltage between the shaft and the hull with the engine not running has always been found to be zero.

In only one ship do we find a measuring value which deviates entirely from the rest, namely where there is an excessive cylinder wear in spite of measured low voltage between the propeller shaft and the hull. We have noted that the cylinder liners were much corroded not only in the upper part but also farther down in the bore.

I cannot find an explanation of this discrepancy at this stage and must leave the finding of the cause to future investigations.

7C. MAIN AND AUXILIARY ENGINES

In some ships of the Broström Lines the cylinder wear, strange to say, has not increased in recent years. This applies primarily to ships equipped with four-stroke main engines, but also a few two-stroke engines have run during the whole time with very small cylinder wear (less than 0.1 mm. per 1000 running hours). The same fuel and lubricating oils have been used as in the other ships.

Referring to four-stroke engines, we have come to the conclusion that the higher temperatures of exhaust gases and cylinder liners enable steam and sulphurous acid to be discharged in the

form of gas through the funnel. Electrolyte is thus not produced in the cylinders of these engines, consequently no electrochemical process will be started and neither will there be any oxidation of the cylinder oil, accompanied by detonation processes.

In two-stroke engines, on the other hand, the surface of the cylinder liners is cooled by the scavenging air, causing both the steam and the sulphurous acid to condense as soon as they strike the cylinder surfaces, thereby starting the destructive process.

Neither have we noted any appreciable increase of the cylinder wear in our two-stroke auxiliary engines. The higher speed of these engines may afford an explanation of the discrepancy. In other words, the available time for oxidation of the lubricating oil may be too short.

8. INVESTIGATIONS CONCERNING DIESEL FUEL OIL

The investigations of the causes of the increased cylinder wear, however, have not been confined to the foregoing factors. In parallel with these tests the fuel oil has been analyzed continuously from 1925 to 1948, and the analysis results submitted to study. I give an account of the observations made in the course of these studies, and supplement this with graphs showing the changes in composition of the fuel oils used for the ships of the Broström Lines from 1925 to 1948.

Fig. 3 shows the cylinder wear in relation to the spec. gravity, viscosity and sulphur content.

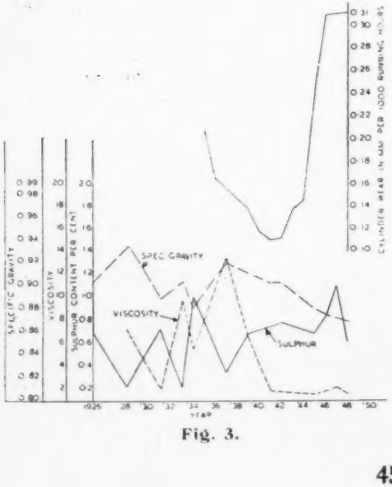

Fig. 3.

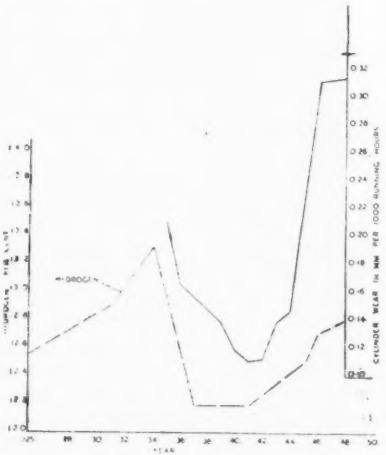

Fig. 4.

The curves indicate the averages and are prepared on the basis of the results of analysis. An examination of the curves does not point to any special agreement with the curve for the cylinder wear.

Fig. 4 shows the cylinder wear in relation to the hydrogen content of the fuel oil. In order to bring out the hydrogen curve we have made use of a diagram prepared by Blackwood and Cloud. In the opinion of other scientists this diagram can be applied to Diesel fuel oils in general.

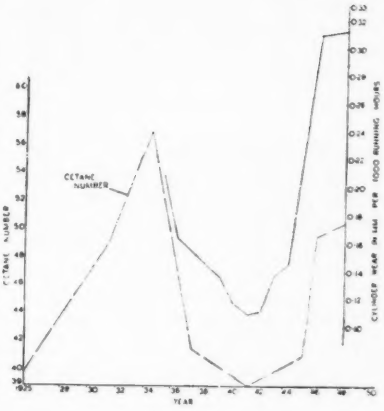

Fig. 5.

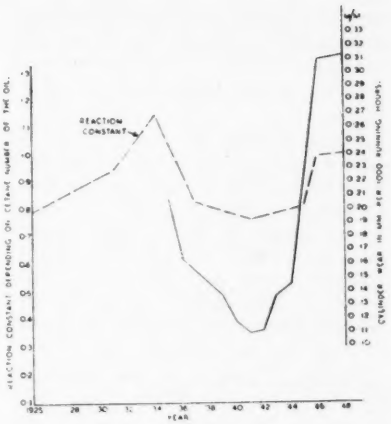
REACTION CONSTANT
REACTION CONSTANT DEPENDING ON CETANE NUMBER OF THE OIL
CYLINDER WEAR IN MM PER 1000 RUNNING HOURS
YEAR

Fig. 6.

From a study of our curve in comparison with the cylinder wear there seems to be a close relationship. The lowest record of the hydrogen curve was made for the years between 1937 and 1941, and shortly after this period the wear also showed the lowest figure. The hydrogen curve then ascends, at first comparatively slowly, whereas the wear curve rises very rapidly. A certain agreement is thus shown although not absolute correspondence.

There is reason to believe that the oil producers in some manner or other changed the processing of the oils after 1935, leading to a reduction of the hydrogen content, which in turn reduced the cylinder wear. This state of affairs would seem to have lasted until the year 1941, when a change of policy might have induced the oil producers to increase the hydrogen content, which has acted unfavorably on the cylinders.

In Fig. 5 the cylinder wear is set in opposition to the cetane number of the fuel oil. (The cetane number of a Diesel oil is expressed in per cent cetane by volume which must be combined with alpha-methyl naphthalene in order to give this mixture the same ignition properties as the Diesel oil.)

The cetane curve is prepared on the basis of the same diagram as used in Graph 4. It will be seen that this curve conforms absolutely to the wear curve, which leads us to suspect that the cetane number is directly correlated to the cylinder wear.

Some change in the processing of the oil took place in 1934 which lowered the cetane number and had a very favorable effect with reference to the wear. In 1941 the curve is at its lowest value with a cetane number of 39. Then there is a slow rise during which the wear increases. From 1945 the cetane curve rises almost perpendicularly in common with the cylinder wear.

Fig. 6 represents the cylinder wear curve in relation to the Diesel oil reaction constant. In a diagram of the cetene number of Diesel oils, prepared by G. D. Boerlage and van Dyck, it is shown that oils with a high cetene number (the cetane number being approximately 13 per cent lower than the cetene number) have a greater tendency to cracking than oils with a low cetene number (understanding by cracking partial dissociation through the influence of heat and high pressure).

On the basis of cetene numbers and

certain research results these scientists reckon with a constant for the oil (the reaction constant) which represents a criterion of the tendency of the oil to crack.

On the basis of this diagram and the known cetane values of our oils we have prepared a graph showing how the reaction constant of the fuel oil has changed in the course of years. There is very close agreement between the reaction constant curve and the wear curve, which in itself points to a certain correlation between the tendency of the oil to crack and the cylinder wear.

In Fig. 7, finally, we have combined the curve for the cylinder wear with curves for the hydrogen content of the Diesel oil, the reaction constant and the cetane number.

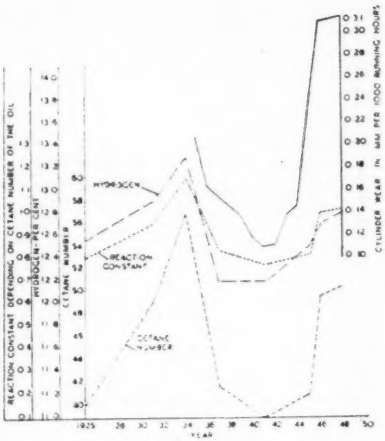

Fig. 7.

A survey is here obtained of the changes of the Diesel oil in these respects in the period under review, and indicates to what extent the curves representing these changes are in agreement with the developments in cylinder wear.

I am ignorant of the causes which led to these great changes in the processing of the oil, but it is to be supposed that different processing methods are applied as a consequence of the technical advances and new substances being obtained from the oil, causing deterioration of the oil in respect of cylinder wear. I trust that if any representatives of the oil companies are present they will give us the benefit of their knowledge of this matter.

9. SUMMARY

With this relation of the conceivable causes of the excessive cylinder wear, based on scientific results, assumptions of the processes involved, experiments, control of the conditions prevailing in the actual running of the engines and aboard ship, and investigations of the fuel oil, I have tried to formulate an opinion that the abnormal wear of the engine cylinders is due to two different correlated causes, which each can appear independently but most often combine to affect the engine cylinders destructively.

The first of these, according to my hypothesis, is composed of three "links" in the destructive process, namely:—

1. *The electrolytic corrosion process* which is produced by the engine being placed under electric tension as the result of low insulation resistance in the electric plant and system on board the ship, and which leads to a more or less severe destruction in the cylinders.

The corrosion test shows that electric tension, if present, greatly increases corrosion of the metal.

Our observation of the correlation between the insulation resistance and cylinder wear in the ships gives us reason to conclude that here is one ground for the deterioration.

2. *Oxidation of the cylinder oil* which is started by the electro-chemical process and results in the formation of pitch and resin-like deposits in the cylinders, thereby diminishing the lubricating properties of the cylinder oil.

The observations which indicate that ships in which the cylinders of the engines are badly fouled have low electric insulation resistance also support our complaints against the electric plants on board.

3. *Uncontrolled detonating combustion,* produced by oxidation products from the cylinder oil, and being possibly the process which has most destructive effect on the cylinder surfaces.

The test with an acetylene-gas flame gave evidence that tensions are produced in a metal if this is exposed to high temperature and ultra-violet light, and the fact that the highest tensions have been found in ships having the greatest wear—ascertained through our measurements—speaks in favor of the conclusion that we have here to deal with a special feature with bearing on the subject.

The second causal connection would seem attributable to the changes in the composition of the fuel oil for Diesel engines.

The correlation between cylinder wear and hydrogen content, cetane number, reaction constant of the fuel oil, and possibly other factors connected therewith, is a question, however, that must be left to oil experts and perhaps other initiated parties with experience in this matter to explain.

I need hardly say that the whole problem of cylinder wear is an enormously complicated one which is intensively studied the world over, and my own ideas on the subject on the basis of the scant material I have been able to submit naturally make no pretense to afford a solution. They may nevertheless contain points of view which may lead to continued investigation and research.

As pointed out by Mr. Christiansson in introducing the subject, we have considered the experience from the operation of the motor ships of the Broström Lines and the results of our investigations interesting enough to afford the groundwork for a profitable discussion of the theories and hypotheses we have ventured to place before the meeting.

Official U. S. Navy Photograph

L. S. M.—Landing Ship, Medium

CORROSION RESEARCH IN THE NAVY

ACKNOWLEDGMENT

MR. JULIUS J. HARWOOD and MR. FRED SCHULMAN, both of the office of Naval Research, Materials and Chemistry Branches respectively, wrote this article. It appeared in the July 1949 issue of *Corrosion*, the official publication of the National Association of Corrosion engineers.

Corrosion has been aptly termed a "major industry in reverse—all loss with never a profit." The annual loss to the economy as a result of environmental and chemical attack has been estimated to be 3½ billion dollars, truly a staggering figure. This drain upon national resources lately has received intensified attention as a result of the experiences of the last war and the necessity for conserving the limited supply of common structural materials.

Corrosion of metals, of which the most familiar example is the rusting of iron and steel, is obviously a problem of national importance. The chemical, metallurgical, mining, power, oil and fuel industries all are vitally concerned. The use of higher pressures and higher temperatures as a result of technological and scientific advances has increased many times the problems of corrosion. Materials which formerly were considered corrosion-resistant for a specified set of operating conditions have become relatively useless as service requirements become increasingly severe.

Of all the industrial and military activities concerned with the destructive effects of corrosion phenomena, the Navy may be considered to be the most seriously affected. The necessity of maintaining the fleet and air arms of the Navy in operational readiness, while they constantly are exposed to the pernicious influence of a natural agent of corrosion (sea water) and confronted with almost every other known type of corrosion phenomena as a result of the multitudinous service operations and processes aboard ships and aircraft, makes corrosion a most serious consideration in the design of equipment and the selection of materials for naval applications.

Compared to the enormous cost of corrosion destruction, relatively insignificant sums have been expended for the scientific study of corrosion. Much of this necessarily has been in applied research, trouble shooting, evaluation and testing, aimed primarily towards the empirical development of corrosion-resistant alloys, protective coatings, corrosion inhibitors and rust-preventives. Relatively few investigations have concentrated on the fundamentals of corrosion. The authors feel secure in saying that the corrosion and surface chemistry program of the Office of Naval Research, in conjunction with related and supporting ONR projects, represents one of the leading corrosion research programs in this country devoted to the understanding of why metals do or do not corrode and the mechanisms and kinetics of corrosion reactions.

The Office of Naval Research, established by an Act of Congress in August 1946, is charged with the overall respon-

sibility for the Navy's scientific research program. For the past three years, ONR has been sponsoring a broad, long-range program in fundamental research covering practically all branches of the physical, biological and naval sciences. This program is executed largely by means of research contracts with academic, industrial, private and government institutions. Equally important functions are the coordination of the scientific research effort throughout the Navy and the dissemination of research information resulting from such endeavors.

The Office of Naval Research corrosion research program, which is jointly administered by the Metallurgy and Chemistry Branches, is aimed at the study and elucidation of the fundamental mechanisms and kinetics of corrosion reactions, and supports the more practical and applied studies underway in the various naval bureaus and laboratories. The object of this article is to describe the ONR corrosion research program and its objectives and to review briefly the progress to date. However, before discussing this program in detail, it may be advisable to describe briefly the prevalent types of corrosion encountered throughout the navy and the general methods currently employed to control them.[1]

CORROSION TYPES DESCRIBED

Corrosion is defined as the destruction of a metal by chemical or electrochemical reaction with its environment. Practically all metals and their alloys corrode, the major difference being the degree and rate at which the corrosion process occurs. The science of corrosion may be traced to the 1830's when Faraday enunciated his now famous Laws of Electrochemical Action, but it was not until the early part of the twentieth century that the electrochemical nature of most corrosion processes was recognized definitely. The establishment of the electrochemical mechanism of corrosion on a quantitative basis is a result of relatively recent fundamental research. Electrochemical corrosion occurs when current flows between anodic and cathodic areas situated at different parts of the metallic surface. Obviously all chemical changes involve electron displacements, but the term "electrochemical corrosion" is reserved for current flow over finite distances longer than inter-atomic spacings. It must be recognized that not all corrosion processes are electrochemical in nature (e.g., simple acid attack), although by far the greatest percentage occurs as a result of electrochemical action. Electrochemical corrosion, resulting from anode and cathode reactions, may be caused by dissimilar metallic contact, impurities and other metallic non-homogeneities, solution concentration effects and differential aeration currents.

TYPES OF CORROSION

The most common type of corrosion, uniform attack or general corrosion, represents the greatest destruction of metal on a tonnage basis. Failure generally is the result of a decrease in section thickness as a result of uniform chemical or electrochemical attack over the entire exposed surface. This type of attack, because of its characteristics, can be controlled by suitable choice of methods described below, and its progress followed easily, thus enabling reliable service life predictions and adequate design thereof. The other types of corrosion to be discussed are more insidious in nature, difficult to predict and often responsible for unexpected and sudden failure.

Intergranular or Intercrystalline

Intergranular or intercrystalline corrosion consists of localized preferential attack at the grain boundaries of metals or alloys. Nickel-chromium stainless steels (e.g., 1808) are particularly susceptible to intergranular corrosion when in a "sensitized" condition because of improper heat-treatment, improper welding procedures, or insufficient carbide stabilization. Many of the brass alloys, extensively used throughout the Navy for condenser systems and numerous other applications, aluminum and magnesium alloys, and nickel, are also susceptible to attack of this nature in many environments.

Pitting

Pitting is one of the most destructive and insidious forms of corrosive attack. Although the overall weight loss might be very slight, failure often will occur because of severe localized penetration. Pitting is recognized readily by the presence of pits due to holes, generally localized in isolated areas.

Chlorides, a principal constituent of sea water, are major pitting agents, which obviously makes this type of attack a major problem to the Navy. However, fresh water also will cause pitting in many metals.

Pitting generally is attributed to a lack of homogeneity in the metal surface. Impurities, scratches, and surface roughness all may promote formation of pits. Pits may form under corrosion products which exist on the metal and also may be promoted by local breaks in "passive films" which form on metal and alloys. Once pitting starts, its progress continues at an accelerated rate. Pitting also may result from impingement or cavitation erosion (described below).

Dezincification

Dezincification, a phenomenon which formerly caused considerable equipment failures aboard ship and is still a problem, usually is observed in the copper-zinc brass alloys. The net result of dezincification is a selective removal of the zinc, leaving as a remainder a weak, spongy, copper structure.

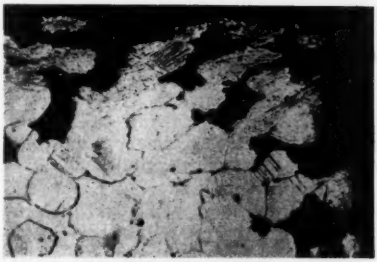

Fig. 1.—Photomicrograph showing preferential corrosion of beta phase along grain boundaries of alpha brass.

The exact mechanism by which dezincification occurs is still a matter of dispute. Two schools of thought exist: one claiming the entire alloy goes into solution with the redeposition of copper and the other upholding the selective leaching of zinc with the copper remaining in situs. Dezincification is observed readily by the change from the yellow color of brass to the distinctive copper color. It may occur as a uniform attack over the entire exposed area or as a plug type attack in localized areas. Dezincification of brasses results in a severe loss of strength and ductility and occurs most often in sea water or aerated liquids. Control of dezincification has been accomplished in some measure by the addition of small amounts of tin, arsenic or phosphorous to the various brass alloys. Lately the copper-nickel alloys have been substituted for the copper-zinc series so this type of corrosion is no longer as troublesome and costly to the Navy as it was.

Erosion-Corrosion

Erosion-corrosion, another major destructive attack encountered in Naval service, is the result of the combination of corrosion and mechanical action. Turbulence, fluid impingement, suspended

solids and other mechanical and abrasive factors greatly accelerate corrosion rates. Some examples of equipment that are especially subject to this type of attack will emphasize the serious nature of the problem: pumps, valves, agitators, pipe lines, heat exchanger tubes and propellers all are susceptible.

If it is realized that most metals and alloys depend upon some sort of protective surface (oxide film, corrosion product, coatings) for corrosion resistance, then it easily is seen that destruction of these surfaces by mechanical action and the continuous exposure of fresh metal will result in a rapid deterioration. One type of erosion-corrosion attack often encountered in naval vessels is impingement and cavitation corrosion.

Impinging streams of water containing entrained gases (bubbles) and cavitation effects, a consequence of the damage resulting from the formation and collapse of cavities in the liquid at a solid-liquid interface, cause particular difficulties with corrosion of ship propellers. Concentration cell effects are associated closely with this type of attack.

Galvanic Corrosion

Galvanic corrosion is the best illustration of truly electrochemical corrosion and is responsible for many of the major corrosion problems. In a broad sense, it includes many of the other corrosion types. The electrochemical contact of two dissimilar metals in an electrolytic medium will result in a potential difference between the two, and a flow of current will occur. Such dissimilar metallic contacts commonly occur where fasteners, bolts, nuts, and screws are used, or when porous metal coatings are applied. The electrochemical reactions occurring at these metal electrodes result in accelerated action at one metal and decreased action at the other as compared to their open circuit corrosion behavior.

The familiar Electromotive Force Series is often useful in predicting the approximate behavior of such reactions. The metals high in the series (base metals-anodic) are less corrosion resistant than those at the bottom (the noble or cathodic metals). However, surface conditions such as oxides, gas films, or other passive films, often cause deviations from "standard" potentials, and

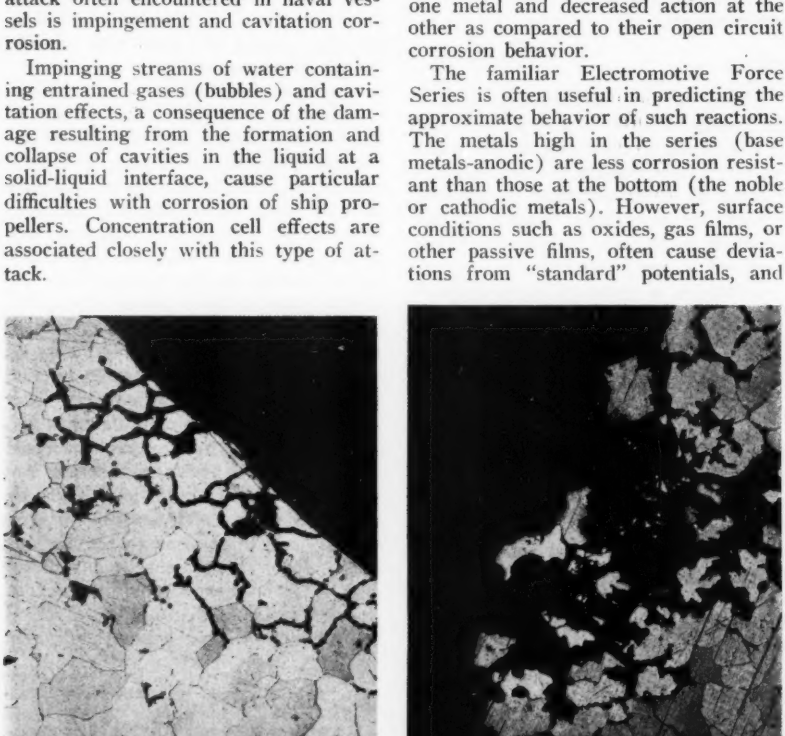

Fig. 2.—Photomicrographs of an aluminum alloy casting showing severe corrosion and penetration. Note the complete separation of grains.

unless the EMF series is intelligently used it may lead to erroneous interpretations.

Galvanic corrosion usually occurs as localized attack near the points of contact and may cause rapid failure though little corrosion is found elsewhere. Galvanic corrosion or couple action also may occur as a result of the existence of local cells due to the presence of impurities, surface inhomogeneities, and concentration cell effects. The principles of galvanic corrosion often are made use of in combating corrosion by cathodic protection (described in greater detail subsequently). The effects of galvanic corrosion may be minimized by using combinations of materials with similar electrode potentials in the medium in which they will be exposed; insulation of joints; ensuring that the noble metal has a smaller area than the metal subject to attack (anode); painting both metals or the noble metal by itself.

Concentration Cells

Concentration cell corrosion, a type somewhat similar to that just discussed, occurs when one metal is immersed in, or in contact with different concentrations of its environment. Localized attack occurs in a manner similar to that of galvanic corrosion. Metal ion concentration cells or oxygen concentration cells, which may exist simultaneously, are both well known types. This type of corrosion often is associated with crevices, scale and surface deposits.

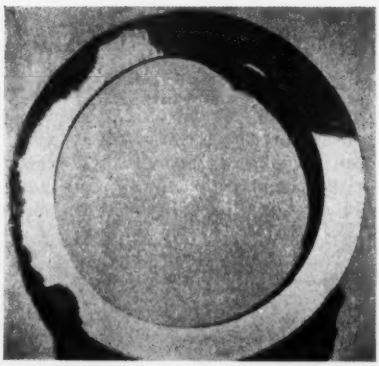

Fig. 3.—A cross-section of a brass pipe which has been buried underground revealing the nature of dezincification. In some areas, it has progressed completely through the pipe wall.

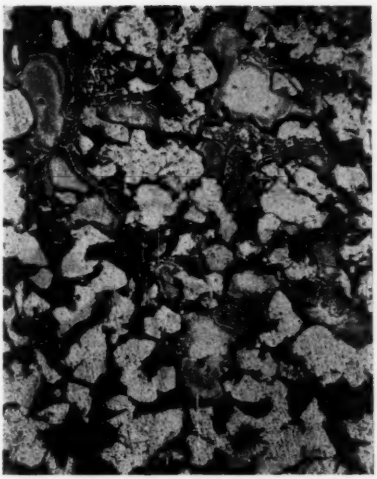

Fig. 4.—Photomicrograph of one of the dezincified areas of Figure 3 showing islands of the original alpha brass in a matrix of porous copper.

Stress-Corrosion

Stress-corrosion or accelerated corrosion resulting from the combination of corrosion and stress, either internal (residual) or externally applied stresses, is probably the least understood of all corrosion mechanisms. The well known

Fig. 5.—Destructive action of erosion-corrosion on stainless steel pump impeller. (Courtesy Dr. M. G. Fontana.).

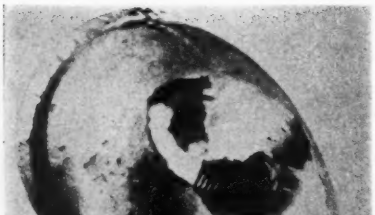

Fig. 6.—Impingement attack on elbow in steam line. (Courtesy of Dr. M. G. Fontana.).

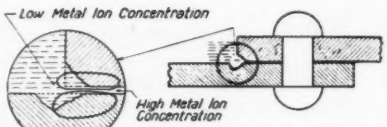

Fig. 7.—Diagrams showing typical condition for concentration cells. (The International Nickel Co., Inc.).

"season-cracking" of cartridge cases and caustic embrittlement of steam boilers are familiar examples of this type of corrosion. Stress-corrosion manifests itself by the formation of cracks along localized paths, generally intercrystalline in nature, although in some alloy systems they may be transcrystalline.

There is no universally accepted mechanism of stress-corrosion although it generally is supposed to be electrochemical in nature because of the existence of microstructural anodic and cathodic areas. These local cells may result from a difference in chemical composition between the grain boundaries and the interior of the grain or from local stress (strain) conditions. High zinc-brasses, aluminum alloys and stainless steels are susceptible to stress-corrosion. Once attack occurs, the presence of stress accelerates the corrosion to a marked degree. Stress-corrosion is a most insidious type of attack. It often is not detected until failure occurs. Factors influencing stress corrosion are degree and nature of stress concentration, nature of corrosion environment, temperature and internal-metal structure. Stress-corrosion often may be prevented by stress-relieving heat treatments, use of protective coatings, surface compressive stresses, cathodic protection and proper design of equipment.

Corrosion-Fatigue

Corrosion-fatigue, another problem extensively experienced in naval applications, is similar to stress-corrosion in that it is a combination of stress and corrosion. The damaging effect of the simultaneous action of corrosion and cylic stresses on metals often is greater than the sum of the individual damage arising from cylic stresses and corrosion. Corrosion-fatigue progresses

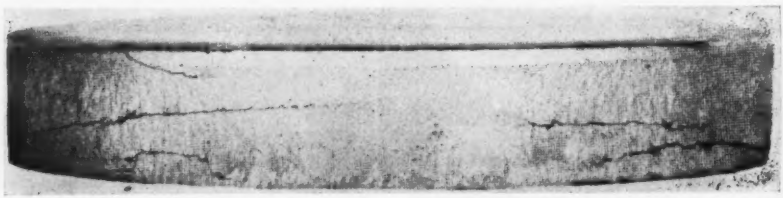

Fig. 8.—Stress-corrosion cracks in brass disk ("season-cracking").

through the formation of deep, sharp pits which become the origin of cracks and subsequent failure.

Often associated with corrosion-fatigue is the rapid corrosion that occurs at the interface between contacting, highly-loaded metal surfaces subjected to slight relative (vibratory) motions known as fretting corrosion. It is characterized by deep pits and surface discoloration. Because it affects loaded surfaces, its effects frequently are catastrophic. As higher stresses in machine parts are employed, frequent trouble from fretting corrosion may be anticipated.

Fig. 9.—Photomicrograph in area of one of the cracks on Figure 8 showing intergranular nature of crack.

High Temperature Corrosion and Oxidation

A type of corrosion which is becoming increasingly important to the Navy is high temperature corrosion and oxidation. As with all other chemical reactions, an increase in temperature results in a similar increase in the rate of corrosion, as well as introducing other concomitant factors. High-temperature oxidation or high-temperature reactions with sulfur and the halogens may, at first glance, seem to be simple chemical combination reactions. However, it now is clearly recognized that oxidation and tarnishing are electrochemical phenomena. Metal ions have been demonstrated to diffuse outward through the oxide (sulfide or halide) film as well as the less frequent diffusion of reactant (gaseous) ions inward. A growing film may be compared to a current-producing cell with the metal-film interface (as anode) supplying cations and electrons for outward diffusion and with the attacking substance-film interface (as cathode) supplying the anions for inward diffusion. The film acts as both the internal and external circuit of a closed cell.

Fig. 10.—Photomicrograph of brass specimen which underwent severe stress-corrosion showing intergranular nature of cracks.

METHODS OF CONTROL

As a result of the destructive effects of the corrosion phenomena described above, numerous methods of corrosion control have been developed empirically to facilitate the practical and satisfactory service application of metals and alloys. The choice of control method to meet a specific corrosion problem is

determined by the economics involved, feasibility of control application and the intended service life. Several methods to control and prevent corrosion are described below.

Cathodic Protection

As previously mentioned, one method extensively utilized for the prevention of corrosion or for minimizing the rate of corrosion makes use of the basic principles of galvanic corrosion. Cathodic protection is accomplished by supplying a flow of current to a metal surface, either by means of a galvanic coupling or by an external current source such that the electrochemical corrosion reactions, operating because of local cells on the surface, either are stifled or neutralized. Cathodic protection thus may be divided into two general classes: protection by the use of sacrificial anodes or protection resulting from impressed currents. Galvanized (zinc-coated) steel is a most familiar example of cathodic protection utilizing the metallic zinc coating as a sacrificial anode. At any break in the zinc coating which is in contact with the corrosion environment, current is generated by the zinc-iron galvanic cell. The current flow is such that zinc is the anode and corrodes preferentially cathodically protecting the steel. Any bare spots on the steel surface will be protected by the surrounding areas of zinc. Sacrificial external anodes of zinc and magnesium are widely used to protect buried pipe lines and hot and cold water tanks. The Navy makes use of similar methods to protect ship and submarine hulls, aircraft and ballast tanks.

An impressed current from an external source accomplishes the same results by neutralizing the local currents produced by the numerous local cells existing on the surface. The applied current enters into the metal over the entire area in a direction opposite to the local cell currents and thereby minimizes corrosion.

Metallic Coatings

In addition to metallic coatings used for cathodic protection, non-sacrificial metallic coatings (e.g., copper or nickel on iron) are often employed to serve as physical barriers to the corrosive environment. However, such non-sacrificial coatings must be well bonded to the base metal, dense and free of pores in order to afford effective protection. If breaks exist in the coating, electrolytic action will occur at the local cells and the current flow will serve to increase corrosion of the base metal, since the latter is anodic to the coating. The type of corrosive medium or environment is most important. For example, aluminum is cathodic to iron in fresh water and can be used as a protective coating only if pore free, but in sea water the aluminum reverses its potential and acts as a sacrificial anode, cathodically protecting the iron. Temperature is also important, since polarity reversals have been noted with temperature changes, zinc being a well known example. Thus, again it is essential to emphasize the limited utility of the Electromotive Force Series in predicting the galvanic behavior of metal couples in various corrosion environments.

Non-Metallic Coatings

Non-metallic coatings, including paints, lacquers, organic and inorganic coatings, serve largely to isolate the metal from the corrosion environment and limit diffusion of both moisture and oxygen. They often are used in conjunction with cathodic protection and inhibitors.

Organic inhibitors are believed to afford corrosion protection by adsorbing on the metal surface and forming a tightly bound mono-molecular film or monolayer of ions. These inhibitors alter the surface potential slightly and function by diminishing the rate at which hydrogen ions or other ions can reach the metal surface. Passivators (e.g., red lead and zinc chromate), on the other

hand, function by making the potential of the metal in contact with the solution more noble. It has been postulated that they strongly adsorb on the metal surface and satisfy secondary valence forces, thus diminishing the reactivity of the metal. Passivators are characteristically oxidizing compounds or strong electron adsorbers. Vapor phase organic inhibitors and rust preventives recently have been developed in an effort to expand Naval use of inhibitors to control corrosion of equipment.

Oxide coatings, ceramic coatings, rubber, glass and phosphate coatings are also examples of non-metallic coatings which are protective.

Deaeration of the liquid environment also has resulted in markedly reducing corrosion losses in boilers, condenser systems and other similar apparatus. Alterations of the corrosion environment by other suitable means may also result in effective corrosion control.

Purification of Metals

Metal purification has successfully improved the corrosion resistance of aluminum and magnesium. The removal of minor detrimental impurities, such as iron and nickel, from magnesium has markedly increased its corrosion resistance such that pure magnesium is virtually unattacked by sea water. Purification reduces the number of local cells and consequently avoids galvanic action. In the case of iron, where the amount of oxygen in solution is important and oxygen concentration cells may be present, purification has not resulted in a great improvement in corrosion resistance to sea water.

Alloying for Chemical Resistance

Alloying for chemical resistance is a popular means for obtaining corrosion control. The applications of stainless steel, Duriron and other corrosion-resistant alloys are too well known to go into further detail. The mechanism of their corrosion resistance, however, is of great theoretical and practical importance. The "passivity" of these alloys is, and has been, a subject of much research and will be discussed further in subsequent paragraphs. Alloying also has been of great value in the development of oxidation-resistant materials capable of withstanding high temperatures.

One other important method of providing control of corrosion is the utilization of adequate equipment design. Proper care in design to avoid crevices and concentration cells, opportunities for stagnant areas, high localized stress areas, hot spots, and to insure proper drainage and surface contouring will do much to retard avoidable corrosion and improve equipment life and operating costs.

It is not to be considered that the corrosion reactions and methods for their control discussed above are unique to the Navy. They are encountered in many other industries and operating environments. The experiences and practices encountered elsewhere have been of tremendous value in assisting corrosion control throughout the Navy and obviously the reverse is also true.

SECTION II

In order that the corrosion projects and problems currently being studied under Office of Naval Research sponsorship may be feasibly discussed, the overall program arbitrarily has been classified into the following general groupings:

1. Passivity and Film Formation Studies.
2. Inhibitor and Passivator Studies.
3. Galvanic Corrosion Studies.
4. General Corrosion Studies, and
5. High Temperature Oxidation and Corrosion Studies.

PASSIVITY AND FILM FORMATION

Most metals and alloys depend upon the formation of protective surface films (passivation) for successful use under corrosive conditions at normal and elevated temperatures. Breakdown of these films results in corrosion failure. It is obvious, therefore, that a knowledge of the mechanisms of passivation and the factors influencing the formation, nature and breakdown of protective films is essential to the complete understanding of corrosion phenomena and the explanation of why metals do or do not corrode. Since the time of Faraday, the phenomenon of passivity has been the subject of a great deal of experimentation and speculation with no completely acceptable theory proposed. The recent issue of the *Corrosion Handbook* defines passivity in the following two ways: "A metal or alloy active in the EMF series is considered passive when its electrochemical behavior becomes that of an appreciably less active or noble metal"; or "A metal or alloy is passive if it substantially resists corrosion in an environment where, thermodynamically, there is a large free energy decrease associated with its passage from the metallic state to appropriate corrosion products."

At the present time two theories of passivity predominate. One, The Generalized Film Theory attributes passivity to the formation of a protective film. Though most proponents of this theory believe the protective film is an oxide layer, others have presented experimental evidence which indicates that adsorbed gases (physically or chemiadsorbed) also may promote passivation. Adsorbed films are thought to result in corrosion resistance by the satisfaction of surface secondary valence forces. The second theory, the Electron Configuration Theory, centers about the fact that most metals which exhibit passivity are those in the transition groups of the periodic table. The atoms of these elements (Cr, Ni, Fe, Mo, etc.) are characterized by incomplete inner shell (d electron) energy levels and by unfilled **d** energy bands in the metallic state. The state of passivity is ascribed to the condition of unfilled **d** bands in the metal or alloy, and the active state to the situation that in effect fills the **d** band with electrons. Thus, adsorbed oxygen or adsorbed oxidizing substances accompany maximum passivity because they are electron adsorbers and do not supply electrons to the metal surface atoms.

The passivity of alloys (e.g. Fe-Cr stainless steels) is attributed to an electron sharing mechanism between the atoms (e.g. iron and chromium) in the lattice. The mechanism for the increased nobility of potential and the decreased reactivity of passive metals and alloys is considered to reside largely in the surface-metal atoms, although in alloys electron interaction (sharing) occurs throughout the metal. However, it is the electron configuration of the surface metal atoms that determines the degree of reaction with a corrosive environment.

Passivation may result from such reactions in accordance with the second definition of passivity stated above due to the formation of protective layers of reaction product. In some cases, adsorption may occur, rather than reaction, which may promote passivation by the mechanism previously discussed. Adsorption or compound formation (reaction) is a function of the ratio of the energies involved in the work function of the metal (a measure of the heat of evaporation of electrons from a metal surface) and the heat of sublimation of the metal. When the work function is small and the ratio of work function to the heat of sublimation is equal to or less than one, adsorption and passivity will occur. When the energy necessary to dislodge an atom from the lattice is

small and the ratio of work function to heat of sublimation is greater than one, then reaction will occur. Metals passive by the consideration of the Electron Configuration Theory may be considered to be covered with a monoatomic or monomolecular film. Thicker films of oxides or other compounds are not included as part of the essential mechanism of passivity. In a broad sense, this theory may be considered as a generalized theory which includes the film theory as a specialized situation depending upon surface electronic configuration.

Recently, an unified mechanism of passivity and inhibition, based on the behavior of local elements in metal surfaces, has been developed.[2] This mechanism suggests that passivity may be achieved either by: a) reduction of the open circuit potential differences between the local anodes and cathodes, b) increased anodic polarization, c) increased cathodic polarization, or d) a combination of these factors.

Under an Office of Naval Research contract, Professor H. H. Uhlig of the Massachusetts Institute of Technology, a leading proponent of the Electron Configuration Theory, is studying the "Effect of Electron Sharing and Intermetallic Compound Formation on the Corrosion of Metals." The purpose of this project is to study the fundamental factors relative to electron-sharing and compound formation in several of the binary and ternary alloys in order to initiate a science of alloying with respect to the chemical properties of metal systems. The study will extend the Electron Configuration Theory of passivity and perform critical experiments designed to test its validity. Two phases of electron sharing have been initiated, namely the investigation of the chemical and physical properties of intermetallic compounds of magnesium (with Group IV B elements) and the study of the chemical properties of adsorbed metal films.

Studies recently have been completed on the compounds Mg_2Sn and Mg_2Sb. Some recent publications[3, 4] report on the measurement of electrical conductivity and thermoelectric power of magnesium-tin and magnesium-lead alloys, together with the compounds Mg_2Sn and Mg_2Pb, as functions of temperature and composition; and the quantitative evaluation and comparison of chemical properties of the alloys and compounds, including the measurement of solution potentials, rates of oxidation at high and low temperatures, rates of reaction in electrolytes, and reaction mechanisms.

It has been shown that Mg_2Sn is an electronic semi-conductor, i.e., the conductivity increases as an exponential function of temperature, whereas the temperature dependence of resistivity of Mg_2Pb is that of a normal metal. Both compounds are more reactive than magnesium when subjected to high temperature oxidation (350°-500°C). The solution potential measurements show that the potential is strongly dependent on the electrolyte. The corrosion reaction, in the absence of air, may be summarized as follows:

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In pure water, single crystals of Mg_2Sn break down into geometric fragments. The rate of breakdown is a function of the hydroxyl ion concentration and occurs in the absence of stress.

The research on the study of Mg-Sn and Mg-Pb compounds is being extended to include Mg-Si and Mg-Ge so that the type of intermetallic bonding may be correlated with chemical and physical properties, especially passivity.

The promotion of passivity by adsorbed metal films is being studied initially with the iron-chromium system. Thin films of iron are deposited on chromium by evaporation and the iron film is then exposed to reagents, such as nitric acid. The residual iron made passive by electron interaction with the underlying chromium is detected either

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by chemical analysis or by use of radioactive isotopes. Precise determination of the thickness of the residual iron passivated by contact with the chromium will be made. The study later will be extended using evaporated films of copper and zinc. It is hoped that such work will give a clear insight into the mechanism of passivity in the stainless and related alloys. Research on the chemical behavior of metal films formed in vacuum and the effect of adsorbed substances, such as gases, on the corrosion characteristics of these films also is underway.

As part of the study of the nature of passivity in the stainless steels, the influence of surface treatment on the variation of potential of 18-8 with time is under investigation. It is believed that interstitial hydrogen may be a factor promoting this time-dependence behavior of the observed potential, and the effect of absorbed hydrogen on the changing potential will be evaluated.

The passivity of stainless steels also is being studied by Professor M. G. Fontana of Ohio State University as part of his ONR program, "Fundamental Studies of Corrosion." The major objective of this research program is the study of the formation, breakdown and nature of surface films formed during the corrosion of metals at normal and elevated temperatures.

The results of these studies, thus far, indicate that 18-8S stainless steel can become passive by means of a physically adsorbed gas.[5, 6] At room temperatures, specimens of this alloy display passivity upon exposure to air and lose passivity upon exposure to reduced pressure (vacuum). This process is reversible in that the alloy can be passivated, broken down, repassivated, broken down and so on, by alternate exposure to air and vacuum. The ready breakdown of the passivity under vacuum at room temperature indicate physical adsorption. Oxides or chemisorbed gases would be considerably harder to remove or break down and would probably require elevated temperatures. Electron-diffraction studies also reveal no oxide film present on the surface. A diffuse and unidentified pattern was obtained which may be a hydrous oxide of nickel or chromium. This material may be of assistance in providing a surface that readily adsorbs gases. It was found that the most effective passivation treatment of all those investigated was exposure to boiling sulfuric acid followed by exposure to air. Air passivation is regarded by Dr. Fontana as the true passivation of stainless steels. Vacuum breakdown of the passive films is pressure and time sensitive. The time for breakdown increases as the pressure increases, and there may be a critical pressure above which the films would not break down. Electron diffraction studies of sulfuric acid-air formed films show no changes taking place in the film as a result of prolonged air exposure.

Interestingly enough, the 18-8 stainless steel also was passivated by exposure to nitrogen, helium gas containing 1.5% nitrogen, and argon. Conclusive demonstration of these effects would provide strong support for this new theory involving a physically adsorbed gas. Gas adsorption studies on metal filings are underway to determine which gases and how much gas can be adsorbed at room temperature.

The passive film formed in boiling, distilled water does not break down under reduced pressure. It was also found that cold worked stainless steels cannot be passivated by the above methods.

Film stripping studies and electrolytic passivation studies have been initiated to study the structure of the films and potential breakdown measurements and erosion-corrosion tests are being used to follow the breakdown of the passive films. Recent work has indicated that potential-time studies recording the characteristics of the specimen while undergoing an actual corrosion test are more informative and applicable to the study of passivation than potential break-

down measurements. Potential-time studies of active and passivated 18-8S stainless steel produced four general types of curves. One represented the active state while the others indicated three degrees of passivity: adsorbed films of a physical and chemical nature and oxide film.

A third aspect of passivity of metals is under investigation by Dr. N. Hackerman of the University of Texas as one phase of this research program on "Exchange Adsorption."

The passivity of chromium has been studied[7] by treating electrolytically deposited chromium surfaces with nitric acid and subsequently oxidizing them in air at various temperatures. Electron diffraction observations were correlated with electrode potential measurements. The conclusions reached were that an oxide film was not the primary cause of the passivity of chromium unless a thickness of the order of a monolayer is considered as a true oxide. It was shown, however, that oxide films can alter appreciably the values of the measured electrode potentials of the metal surface, although these noble potentials obtained with oxide films need not be identified with passivity. It appears that the rate of decay of the noble potential can be used in considering the "degree of passivity." Thus the potential changes rapidly to a more basic value for the less truly passive metal.

No evidence was obtained to indicate that an "oxide film" of monomolecular dimensions could not explain the behavior of the metal. The Electron Configuration Theory also explains the behavior of chromium satisfactorily, since this theory can be shown to account for the existence of a monolayer.

Dr. Hackerman also is studying the effect of chloride ions on passive metals. The extent and rate of association, or attachment, of chloride ions from aqueous solution on chromium and on stainless steel surfaces is being investigated by the use of radioactive chloride ion. It is well known that passive metals become active toward corrosion in the presence of halide (particularly chloride) ions but the actual mechanism of chloride induced corrosion still is not understood. It had been postulated that chloride ion catalyzes the breakdowns of the electronic configuration of the surface metal atoms at certain local areas causing galvanic action to occur. Results of this study, thus far, suggest the possibilities of a chemisorbed layer of chloride ions, and indicate the importance of the chloride ion attachment as a factor in the destruction of passivity of chromium. The visible corrosion of the stainless steel samples increased with the amount of chloride ion on the surface as measured by "counting" techniques, and most extensive corrosion was observed in the most concentrated radioactive areas. Further work on this phase of the program is continuing.

The adsorption of gases on low carbon steel, stainless steel and chromium also is being studied to extend the existing knowledge concerning passivity by increasing both the number of metals and gases investigated. Isotherms of oxygen and chloride adsorbed on 1020 steel, stainless steel and chromium powder have been obtained, and those of carbon monoxide and the oxides of nitrogen currently are being determined. Isotherms were obtained on the powders after either baking, reducing with tank hydrogen, or reducing with deoxygenated-hydrogen. The results, thus far, show general trends of increased adsorption in the order of chromium, stainless steel, 1020 steel and baked, plain hydrogen reduced and deoxygenated-hydrogen reduced surfaces. In the case of chlorine, indications are strong that chemisorption occurs.

Experiments are underway to determine the exchange adsorption of chlorine gas on an oxygenated (passivated) chromium surface. Thus, the extent to which chlorine can replace oxygen and break the passive film will be determined.

INHIBITORS AND PASSIVATORS

The importance of inhibitors in preventing corrosion of metals is demonstrated by the widespread applications in metal cleaning, metal finishing and metal protection processes. It is agreed generally that corrosion inhibition by organic molecules is the result of the inhibitor molecule forming an adsorbed film over the metal which prevents penetration by corrosive agents and, therefore, causes the surface to be less susceptible to corrosive attack. However, the mechanism of the adsorption process, the effect of inhibitor molecule size and composition, and conditions surrounding the adsorbed layer in corrosive media are still unknown to a large extent. These factors are being studied by Dr. Hackerman, under the above contract, in a two-phase program of inhibitor action. The investigation of "The Adsorption of Polar Organic Compounds on Steel Surfaces" is concerned with adsorption of polar organic molecules (organic acids, alcohols, esters and amines) from a non-polar solvent, benzene, on metal surfaces which are relatively planar. Much of the past experimental work has been done on porous, high surface area solids. A method of measuring relatively small amounts of adsorption on planar surfaces has been developed and preliminary studies have been completed. A test to determine the effectiveness of various compounds as inhibitors has been devised and is being applied to all compounds studied.

Another phase of this program is the "Study of Adsorption of Inhibitors from Solution" by means of an EMF method. Dr. Hackerman has postulated that adsorption of corrosion inhibitors takes place over the entire surface of metals, thus reducing the galvanic corrosion due to local anode and cathode areas.[8, 9] The adsorption of positive cationic inhibitor materials (e.g., amines) results in equipotentialization of the surface by causing a displacement of electrons from the metal (anodic area) toward the positively charged particle rather than toward the cathodic area and provides a diffusion or resistance barrier. To demonstrate this, time-potential curves at various inhibitor concentrations on anodically and cathodically polarized specimens as well as on unpolarized specimens were obtained. A study of the adsorption of amine inhibitors on copper and steel specimens indicated that adsorption at anodic spots actually may cause a decrease in solution tendency, whereas adsorption at cathodic regions probably inhibits by means of diffusion, resistance changes or overvoltage effects. Continued investigation of the "partial oxidation" theory is underway to further the elucidation of the nature of the adsorption processes on metals from solutions.

The mechanism of inhibitor action afforded by the "passivator" group of compounds is being studied by Dr. Uhlig as part of his passivity research program. According to Uhlig[10] passivators usually are oxidizing agents which produce their effects by changing the potential of the base metal in a noble direction instead of producing merely a physical barrier to corrosion. The corrosion protection offered by chromates, hexametaphosphates, nitrites and other passivators is being investigated to gain a further insight into the true nature of passivator action. Corrosion rate studies, potential measurements and surveys making use of radioactive isotopes are being employed.

The investigation of the "Role of Inhibitor Films in Electrode Processes" was conducted by Dr. H. J. McDonald of the Illinois Institute of Technology. The chief objectives of this program were: a) to obtain polarization data on a number of representative metals, using anodic and cathodic inhibitors and, from the curves, determine if a scale of inhibitor effectiveness could be formu-

lated; and b) to study the polarization characteristics of corrosion accelerators and contrast their behavior with that of corrosion inhibitors.

Two instruments, the Pulse Polarizer and Scope Polarizer, were developed for the quantitative evaluation of inhibitors by determining their effect on the electrode polarization of metals. The Pulse Polarizer employs a system of electronic circuits to polarize an electrode over a brief time interval. The polarization and depolarization at the surface of the electrode are recorded continuously on a high-speed strip chart and show up as a curve which is distinctive for each set of conditions. The polarizing circuit consisted of a condenser (125 mfd) which is charged to 310 volts. Discharge of this condenser brings about the polarization at the electrode surface. The polarization potential is amplified by means of an electronic DC voltage amplified and then recorded.

Both anodic and cathodic polarization may be studied with the apparatus, but for inhibitor studies, cathodic polarization is of particular interest. The voltage applied is constant for each pulse, and the time interval during which the surface is polarized is between 0.05 and 0.1 sec. When a pulse is applied, the potential first drops to a new value corresponding to the polarization under extremely high current density, then the curves return toward the normal potential at various rates. A test usually is complete within several minutes. More detailed descriptions of the Pulse Polarizer and the results of the studies of a large variety of inhibitors have been reported.[11, 12, 13, 14] It was found that the Pulse Polarizer is useful in establishing a relative scale of inhibitor effectiveness, but that complete interpretation of the polarization curves often was difficult.

In view of the difficulty of theoretical interpretation of the polarization curves, a study was undertaken on the "Influence of Surface Films on the Electrodeposition of Hydrogen on Palladium" in order to obtain a clearer understanding of the factors governing polarization phenomena and inhibitor effectiveness. It was found that the transmission of hydrogen through a palladium cathode depended largely upon the nature of the adsorbed surface film. Metallic films, severely reduced the transmission of hydrogen, whereas adsorbed organic films might also cause an increase in the transmission depending upon the adsorption behavior of the compound and film thickness.

GALVANIC CORROSION

As previously mentioned, galvanic corrosion is responsible for many major corrosion problems and generally is caused by dissimilar metal contacts or concentration cell effects. The subject of bimetallic corrosion has received considerable attention in the past, but it is still difficult to predict accurately the result of placing two metals in contact under a particular set of environmental conditions. The importance of environment often has been overlooked, and little is known yet concerning the kinetics of the cathode reaction which is a corrosion-controlling process. In order to gain more insight into the complicated phenomena which occur during bimetallic corrosion, Dr. R. T. Koenig, of the University of Colorado studied "Bimetallic Corrosion in Neutral Electrolytes" under Office of Naval Research contract. Two important factors influencing the amount and distribution of corrosion in bimetallic couples in neutral electrolytes were investigated:

1. the relative areas of anodic and cathodic metal; and
2. the varying composition and concentrations of salt solutions.

Partial and total immersion tests with numerous metallic couples using differ-

ent concentrations of sodium chloride and calcium chloride solutions are underway. These tests are being supplemented by electrochemical studies of the oxygen depolarization reaction occurring at the various metal surfaces. Simple specimen shapes, utilizing coupling by means of threaded connections or by ring-plug combinations are being employed. The specimens are free from local strains and avoid the high electrical resistance generally encountered when external electrical connections are used for coupling. Anodes of zinc and iron have been coupled with copper, brass, stainless steel and other metals, and many data have been accumulated. It appears, thus, far, that the observations made differ markedly from those that have been reported previously in the literature. For example, it was found that under conditions where hydrogen evolution is of minor importance in determining the corrosion rate, there is no noticeable effect of the actual junction of two metals upon the distribution or intensity of the resulting corrosion. A report summarizing the results of the investigation will be available in the near future.

The investigation and explanation of the phenomena occurring when a metal is placed in electrolyte, containing no ions of the exposed metal, is the objective of the research program at Clark University, under the direction of Dr. D. MacGillavry, Jr. Little systematic research on this general problem, the nature and behavior of metal-solution phase-boundary potentials when no common ion is present, has been conducted and no adequate theory to explain the phenomena exists. These are extremely important in the corrosion of metals under practical conditions.

The research is in its early stages and it is planned, first, to develop techniques for the measurement of phase-boundary potentials; second, to determine how these potentials depend on a number of experimental factors, especially temperature, composition of the liquid phase, and composition of the electrode in case the metal is an alloy; and third, to use the experimental data as the basis for the theoretical explanation of the phenomena.

GENERAL CORROSION STUDIES

At the University of Oregon, Dr. P. Van Rysselberghe is studying corrosion phenomena using polarographic techniques. The polarographic method invented about 1925 by Jaroslav Heyrosky, is based on the interpretation of the current-voltage curves that are obtained when solutions of electroreducible or electrooxidizable substances are electrolyzed in a cell in which one electrode consists of mercury falling dropwise from a very fine bore capillary glass tube.[15] From the unique characteristics of such current-voltage curves both the species and concentration of the electroreducible or electrooxidizable substances present in the solution can be determined. This sensitivity of the polarograph to minute concentration charges of reducible substances provides the basis of rapid and sensitive corrosion tests by following the consumption of oxygen, carbon dioxide, etc., by metallic surfaces dipped in electrolytic solutions containing these agents.

In Figure 11 is shown a schematic arrangement for measuring current-voltage curves by the dropping mercury electrode, the polarograph. In this diagram, **A** is the electrolysis cell containing the solution to be analyzed, **B** is the dropping mercury, and **C** is a stationary pool of mercury on the bottom of the cell which serves as the second electrode. The dropping electrode consists of a drawn out capillary tube (whose internal diameter at the tip is about 0.03 mm) connected to a reservoir of mercury. Mercury drops issue from the capillary at the rate of about one drop

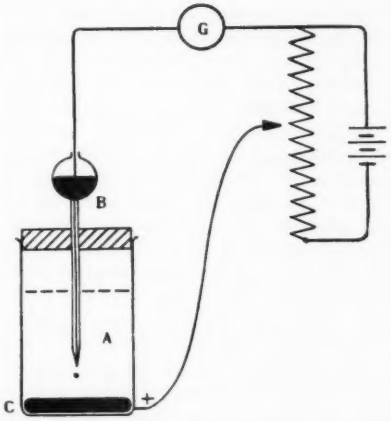

Figure 11—Schematic drawing of polarograph.

every 2 to 4 seconds. The drops are very small, having a maximum diameter at the breaking point of about only 0.5 mm. The cell is connected in series with a calibrated galvanometer, G, to a battery and rheostat, by means of which an electromotive force of the battery can be applied to the cell. The current-voltage curves are obtained by gradually increasing the applied electromotive force and noting the current indicated by the galvanometer. The current ordinarily is quite small, seldom exceeding 50 microamperes. A typical current-voltage curve, or polarogram of oxygen is shown in Figure 12. The potential at which the curve begins to show a rapid increase of current with a small increase in voltage is called the decomposition potential. The midpoint of the vertical portion is called the half-wave potential and the horizontal portion is the limiting current. The decomposition potential is characteristic of the substance and the height of the curve is a measure of its concentration. The first current break is caused by the reduction of oxygen to hydrogen peroxide and the second, by the reduction of the hydrogen peroxide.

It was established that percarbonates can form during the reduction of oxygen in the presence of carbon dioxide,[16] and on the basis of these studies the investigation of the consumption of oxygen, carbon dioxide and their mixtures by various metals and alloys was undertaken. The technique was to expose the metal in a salt solution containing known amounts of carbon dioxide or oxygen or both and by the use of the polarograph to determine the rate of disappearance of these gases. Polarograms of these tests indicated the importance of hydrogen peroxide formation during corrosion of zinc and aluminum and the influence of carbon dioxide in retarding the corrosion of these metals. Further research on the role of hydrogen peroxide and percarbonic acid in corrosion processes and on the application of polarographic techniques to the study of the rates of corrosion of metals currently is underway.

The "Fundamental Studies of Corrosion" program at Ohio State, in addition to the passivity studies discussed previously and the high temperature film studies to be described later, also encompasses studies on the intergranular corrosion of stainless steels, including the effects of stabilizing and welding treatments, and studies on the corrosion

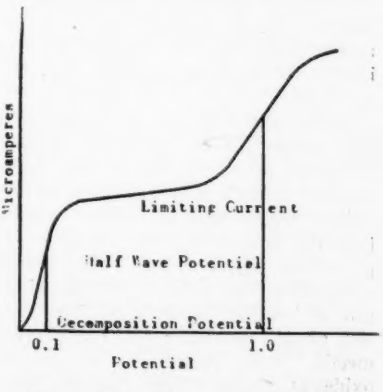

Figure 12—Oxygen polarogram.

of copper and copper base alloys. The latter study will investigate the influence of velocity, oxygen concentration, temperature and time on the corrosion and "passivation" of copper and copper base alloys in various media. As an initial step, the influence of these factors on the single electrode potential of copper in dilute sulfuric acid is being studied to determine the role of copper ion and the oxygen concentration cell in the corrosion of copper. Future research subjects include the application of radioactive tracers in fundamental corrosion studies; investigation of the corrosion of titanium and its alloys; thermogalvanic corrosion; mechanism of internal oxidation and others.

At the University of Utah, Dr. George Hill is studying the "Corrosion of Metals in Contact with Liquids" in the light of the theory that the role of various gases and salts dissolved in water is largely that of providing ions for the conduction of electric current. The validity of this theory and the determination of the specific effects of particular substances are being investigated by studying the influence of varying concentrations of SO_2, Cl_2, H_2S, CO_2, K_2CrO_4, etc., on the rate of corrosion of pure metals and alloys. The data will be interpreted in terms of Eyring's "Theory of Absolute Reaction Rates" which will give the free energy, heat and entropy of activation of the rate determining step, and an elucidation of the mechanism of the reactions.

HIGH TEMPERATURE OXIDATION

The corrosion and oxidation of metals and alloys at elevated temperatures are rapidly becoming problems of major importance to the Navy. The performance of rockets, jet engines, guided missiles, etc., is dependent primarily upon the development of oxidation-resistant materials capable of withstanding extremely high temperatures and the erosive effects of hot gases. In this field also, the Office of Naval Research is supplementing other alloy development programs with several fundamental research programs aimed at the better understanding of the mechanisms and kinetics of high temperature oxidation processes.

Metals generally can be divided into two groups for consideration of their oxidation products. One group, composed of the lighter metals, forms porous oxides, which have a smaller volume than the equivalent metal consumed in producing these oxides. These metals generally exhibit a linear rate of oxidation at constant temperature. The second group, aluminum and the heavier metals, form protective, non-porous oxides of greater volume than the equivalent metal consumed. Many metals of this group exhibit a parabolic oxidation rate (inversely proportional to the square root of time). Under some conditions, the oxidation process is proportional to the logarithm of time.

Factors influencing the rate of oxidation and nature of the oxidation product are orientation of the base metal, lattice structure of the metallic substrate, metal composition, surface conditions and environment, and temperature. Although oxidation usually is considered a direct chemical reaction in which the rate is controlled by simple diffusion of the reacting atoms through a non-porous corrosion product film, recent work has demonstrated conclusively that oxidation may occur by metal ions diffusing outward through the film. This electrolytic theory of oxidation compares the growing film to a current-producing cell in which the metal-film interface acts as the anode supplying cations and electrons for outward diffusion and the environment-film interface acts as a cathode supplying the anions for inward diffusion. The film acts as both the internal and external circuit of a closed cell.

At the Illinois Institute of Technology, Dr. D. D. Cubicciotti is studying the "Determination of the Kinetics of Reactions of Metals with Gases at High Temperatures." The direct measurement of the rates of gas-metal reactions is being made by means of a gas volumetric method with gas circulation under atmospheric or low pressure. Microbalance techniques are being employed as an auxiliary method for equilibrium measurements and thin sheet investigations.

One of the initial phases of this program (which formerly was under the direction of Dr. H. J. McDonald) was the study of electropotentials in growing halide and oxide layers on metal.[17] An electrode probe was developed which enabled the experimental determinations of these electropotentials and the subsequent calculation of the ionic transference numbers. The probe method proved to be a sensitive tool, responding to all factors which influenced the transference numbers. The oxides on iron, stainless steel, nickel, aluminum, zinc, molybdenum and tungsten were found to be predominantly electronic (cationic) conductors. In anionically conducting scale, where the zone of growth was at the scale-metal interface, previous criteria were found to be inadequate.

A study of the zone of metal phase consumption (locus between the initial surface of the metal and the surface of the retreating metal phase) revealed that even in the cases when a scale grows by outward diffusion of metal or metal ions, the scale also must grow on the other side towards the metal.[18] The conditions of scale growth on both sides of the scale are dissimilar; consequently, the structure of the scale tends to be different in both layers formed. A process of cavity and column formation and exchange of oxygen in the zone of metal phase consumption was postulated. It also was noted that as a consequence of secondary processes, disassociation and recrystallization may be responsible for a third layer formation.

Current work under way includes the study of the effects of ionization of gases on oxidation rates at high temperatures and the determination of the rates of reactions of zirconium and beryllium with oxygen, nitrogen and oxides of nitrogen.

As previously indicated, one phase of the "Fundamental Corrosion Research" program at Ohio State University is concerned with the nature and protective value of surface films formed on metals and alloys at high temperatures. The composition and structure of these films are being investigated by chemical and metallographic examination and by electron-diffraction studies. The mechanism and rate of formation, limiting temperatures and other pertinent factors are being investigated. Such information is useful for prediction of the service life of alloys operating at high temperatures and is valuable for the scientific development of improved alloys.

Several of the stainless steel varieties have been exposed to varying oxygen atmospheres for different temperatures and time treatments, and the oxidation process followed by weight measurements and diffraction studies. Although considerable data have been accumulated, the analysis has not reached the point where definite conclusions have been drawn. Over short periods of time, for a number of the steels, oxidation proceeds in accordance with a logarithmic rate, changing to the parabolic relationship with longer exposure time periods. One interesting observation of some preliminary studies of 18-8S at 1800° F. was that oxide films formed in low-oxygen atmospheres are more protective to further oxidation than scales formed in higher-oxygen atmospheres. This indicates the possibility of developing pre-induced films by pre-treatment for increased oxidation resistance.

Somewhat similar studies are being conducted by Prof. H. J. Yearian at Purdue University under Project SQUID. The objective of this program

is the study of corrosion in connection with jet propulsion. The purpose of the research is to identify the corrosion products and to investigate the process of corrosion and its dependence on the chemical and physical properties of the materials and on the conditions of exposure. The formation, structure, properties, and resistive function of oxides, formed in high temperatures, and gaseous oxidation of ferrous alloys (in air and oxygen atmospheres) are being investigated.

Electron diffraction techniques are being used to study thin oxide films and the surface of thicker oxide layers. The diffraction patterns are obtained at temperatures of oxidation by means of special instrumentation. The physcial arrangements of the oxides in the scale layers will be studied by optical microscopy. X-ray-microradiography and photoelectric image formation methods. Electron microscopy will be used to study oxide formation relative to grain boundary precipitation of carbides and/or nitrides in high chrome steels.

The high temperature oxidation products formed in oxygen and in air on chromium steels, ranging from 5 to 26 per cent chromium content, have been analyzed by X-ray diffraction. A regular variation of phases present can be correlated with change in oxidation resistance. In contrast with pure iron, no FeO was found in the oxidation of iron-chromium alloys, indicating a possible function of chromium in producing an oxidation resistant material. The resistance to oxidation generally decreases as the chromium content of the alloy is diminished and as the temperature is increased. A limited amount of work on the air oxidation of nickel-chromium steel also has been completed.

These oxide "existence" studies in oxygen atmospheres at various temperatures and times, using X-ray analyses of bulk oxides, are continuing to determine more definitely the correlation between oxide structure and oxidation resistance.

Since it is recognized that the protection afforded a metal by its oxide coating may not be due to the bulk oxide acting as a mechanical barrier separating the reactants, but rather to some special properties of a relatively thin layer, presumably next to the metal, a considerable portion of the research is being directed toward the investigation of the formation, structure and properties of thin oxide films, using electron diffraction and electron-microscope techniques. For the diffraction studies, oxidation is being accomplished *in situ* in the diffraction camera.

Film stripping techniques also are being utilized to examine thin films by transmission electron diffraction. The electrical properties of the protective oxide layers, as a function of composition and temperature also will be investigated.

The diffusion mechanisms operating during oxidation processes are being studied as part of a large comprehensive program at Carnegie Institute of Technology on the "Kinetics of Reaction in the Solid State" under the direction of Dr. R. F. Mehl. This program is making extensive use of radioactive tracers in the study of diffusion processes, since the techniques are extremely adaptable to the study of diffusion mechanisms in oxide films, thereby allowing the determination of the mechanisms of the scaling of metals. It is possible thus to determine whether it is iron or oxygen which diffuses through the oxide layer during the scaling of iron. A thin layer of radioactive iron is electroplated on a polished surface of pure iron and subsequently oxidized under controlled conditions. The distribution of radioactive iron through the scale indicates the nature of the diffusion process.

Initial results already have indicated that the generally accepted mechanism of oxidation of iron is not sufficiently adequate and it is hoped that information derived from these studies, together with kinetic data now available in the literature will permit a complete recon-

sideration of the mechanism and kinetics of the scaling of iron. Future work will include the oxidation of copper and other metals.

It is well known that most metals, including the common heat-resisting alloys suffer a progressive deterioration in strength and in ductility when they are stressed at high temperatures for a considerable length of time. Whether deterioration is due solely to the effects of time, temperature and stress or whether atmosphere reactions are an important cause of deterioration has not been investigated thoroughly in the past. There is some evidence in the literature that atmosphere reactions have an important effect on high temperature deterioration of metals. For instance it has been shown that, during stress-rupture tests of steels in oxidizing atmospheres, the slope of the rupture-time curve (which is a straight line on a log-log plot) changes abruptly after a finite test time. The change of slope occurs at the same time as intergranular oxidation occurs, and generally the fracture changes from a ductile transcrystalline failure to a brittle intergranular type. The same metal tested in an oxygen-free atmosphere does not exhibit these characteristics. The effect of atmosphere reactions on heat-resisting alloys, a potentially important problem in high temperature power plants, has been investigated only to a limited extent and the results are not conclusive.

In order to investigate these effects, Prof. O. C. Shepard at Stanford University is studying the "Effects of Atmosphere Reaction on the Deterioration of Metals at Elevated Temperatures." This program first will determine whether atmosphere reactions have an important effect on the stress-rupture properties of high-temperature alloys and then by the use of simple, pure metals determine the mechanism and nature of these effects. Tests are being conducted in inert atmospheres and various environmental atmospheres in which heat-resisting alloys commonly are used.

Results to date indicate atmosphere reactions have a definite effect upon the high-temperature strength properties of metals and alloys. Low-carbon steel specimens subjected to an inert, purified helium atmosphere, exhibited greater and more uniform ductility and greater rupture life than specimens tested in atmospheres containing hydrogen, water vapor, oxygen and air. The latter specimens exhibited brittle failure with numerous surface cracks. Confirmatory tests are underway. The program also will include the study of the effect of strain rate on the embrittlement of pure metals subjected to stress-rupture tests in active and inert atmospheres at elevated temperatures, in order to determine whether the reduction in ductility is due to the environment or the effect of low strain rates and high temperatures.

A discussion of the above presented corrosion research program would not be complete without consideration of the support provided not only by the closely allied fields of study within the Metallurgy and Chemistry Branches of the Office of Naval Research, but by many other research projects under way throughout the other scientific divisions of ONR. Particular emphasis must be made of the inter-relationship to corrosion of the research programs in such fields as electrochemistry, metallurgy, adsorption, surface reactions, diffusion and other solid state reactions. Much fundamental information will be forthcoming from these programs which will aid in the understanding and control of the numerous corrosion processes encountered in day-to-day operations. To cite a specific example of the cross-fertilization of research programs, it may have been noted that there were no projects in the current corrosion research program specifically directed towards the study of stress-corrosion cracking of metals and alloys. However, several metallurgical and chemical research projects now underway will make definite contributions to the understand-

ing of the mechanisms of stress-corrosion cracking. Studies on diffusion, flow and fracture of metals, grain boundary behavior and numerous phases of the above corrosion projects are accumulating knowledge which will aid in interpreting the stress-corrosion behavior of metal systems. It might be said that as a start toward the problem, independent investigation is under way of the fundamental nature and laws of corrosion and the behavior of metals under a state of stress, which ultimately can be combined for the solution of stress-corrosion problems.

It must be borne in mind that this report does not include the large amount of corrosion work underway in the many laboratories of the material Bureaus and the Naval Research Laboratory. Space does not permit a description of the extensive efforts made by these activities to solve important specific naval corrosion problems.

In conclusion, the authors desire to point out that in their presentation of the corrosion research program of the Office of Naval Research, no attempt was made to present all significant findings of each project. Those readers who might be interested in details of a specific project are referred to the list of references appended to this paper. Future progress of the studies described above will be reported similarly by the project investigators in the usual scientific journals.

We hope the above summarization will provide an adequate picture of the research undertaken by the Office of Naval Research to further the understanding of why metals do or do not corrode and to study the mechanisms and kinetics of corrosion reactions so that predictions may be made of the behavior, towards corrosion, of metals and alloys under any environmental conditions likely to be encountered by the Navy. It often is pointed out that the Navy, despite the development of passivators, inhibitors, rust preventives, corrosion-resistant alloys, protective coatings and whatnot still depends upon the old method of chipping hammer and paint brush for corrosion control. It is hoped that results of this and other similar programs will provide the necessary information to afford scientific control over the ubiquitous processes of corrosion and to insure unfailing corrosion protection.

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Official U. S. Navy Photograph

An existing design of ship (Landing Craft Infantry, LCI) is adapted to a new weapon—The Rocket—World War II.

PRELIMINARY REPORT OF DEVELOPMENTS IN INTERRUPTED SURFACE FINISHES

ACKNOWLEDGMENT

The 1949 Proceedings, Vol. 161, of The Institution of Mechanical Engineers, contained this article by MR. L. S. MARTZ, of the Micromatic Hone Corporation, Detroit, Michigan. The article discusses the relatively unexplored field of interrupted surface finishes as an approach to bearing efficiency. Service use and laboratory tests of three generic types are reviewed: mechanically, chemically, and electrolytically generated surface finishes and their combinations.

INTRODUCTION

The story of mechanization is essentially the story of the mechanical bearing: "a part in which a journal, gudgeon, pivot, pin, or the like, turns or revolves" (Webster). The functional operation of these two parts requires a third element, a lubricant. These three elements of a bearing couple have partially inherent and partially developed properties, which, under widely varying conditions of production methods and use, combine to establish "bearing quality." The inherent properties comprise metallurgical combinations of chemical and physical qualities; the developed properties comprise the resultants of metal working production methods, design, and service factors, and alteration of all these properties under load-speed-temperature conditions of operation.

It is common practice to consider plain bearings in two usage classifications:—

(1) Boundary lubrication bearings, generally including all internal combustion engine cylinder bores and component parts, and others.

(2) Bearings intended to operate under hydrodynamic full fluid film lubrication, including slow-speed, heavy-duty, and high-speed types for both uni-directional and rotating loads.

In recent years it has become increasingly obvious that increased load-carrying capacity, life, and operating speed of plain bearings must be based on better understandings of the need for establishing more rapid and dependable compatibility between the two bearing parts and the bearing lubricant. Under development pressures bordering on desperation, practically every conceivable combination of bearing materials has been tried out in some form of application. Efforts to standardize the catalytic quality of oil lubricants through additions intended to contaminate the bearing surfaces have been at least partially successful. With all this, however, it is obvious that other important relationships are involved in the successful operation of a plain bearing.

An existing design of ship (Landing Craft Infantry, LCI) is adapted to a new weapon—The Rocket—World War II.

To many people it seems evident that proper implementation of the wedge-film theory of lubrication (Kingsbury, 1931) for present and future higher speed, higher output applications will require some amplification of present concepts of bearing design, bearing part production methods, and lubrication control. Recently an increasing number of investigators have been turning with renewed interest to the unexplored field of interrupted surface finishes as an immediate approach to increased bearing efficiency.

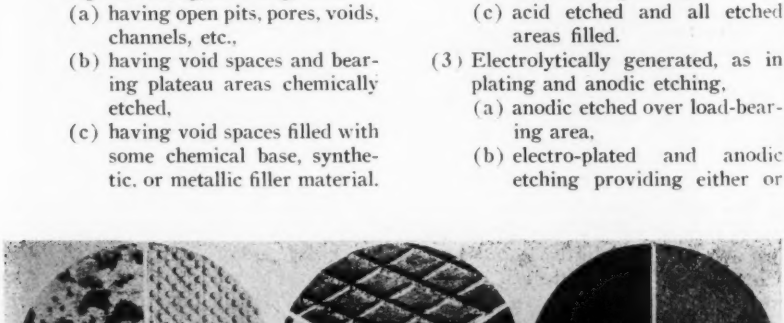
GENERAL CONSIDERATIONS

The present status of the development work that has been done to date may be best considered in three generic classifications:—

(1) Mechanically generated, including machining, knurling, etc.,
  (a) having open pits, pores, voids, channels, etc.,
  (b) having void spaces and bearing plateau areas chemically etched,
  (c) having void spaces filled with some chemical base, synthetic, or metallic filler material.

(2) Chemically generated, as with acid etching,
  (a) acid etched all over load-bearing area,
  (b) acid etched with mask over load-bearing areas,
  (c) acid etched and all etched areas filled.

(3) Electrolytically generated, as in plating and anodic etching,
  (a) anodic etched over load-bearing area,
  (b) electro-plated and anodic etching providing either or



(*a*) (*b*) (*c*)

Fig. 1.—Widely Variable Physical Characteristics of Three Basic types of Interrupted Surface Finishes.

(*a*) Pitted, having isolated pits, regularly or irregularly disposed, surrounded by connected plateau area, as in knurling, mask etching, etc.

(*b*) Channelled, having isolated plateaus or peaks, surrounded by continuous or connected channels, as in machining, scribe and etch, or electro-plating and electro-etching.

(*c*) Porous, having a sponge-like structure of the surface metal as in most chemical etching and to some degree in electro-etching.

both, pits and sponge structure,

(c) electrolytically generated and filled.

Each of the above basic methods produces some one of three types of interrupted surface finishes, as shown schematically in Fig. 1.

Any one or all of the above described methods may be used to develop a microroughness surface effect for only the initial run-in period until the compatibility of the surfaces is obtained; or may be used to produce deeper or so-called permanent relief of the surface to last for thousands of hours of operation.

Because open pits or voids cause high oil consumption in engine cylinders, and because it is generally believed that hydrodynamic lubrication requires the maximum uniformity of wedge film, consideration of interrupted surface bearing areas must necessarily include the use of fillers for these applications.

MECHANICALLY GENERATED INTERRUPTED SURFACES

The concept of mechanically generated ridges and recesses or grooves as a means of permitting a bearing to conform itself to a journal quickly, to produce "a steady uniform rotation without undue friction," and to afford adequate lubrication for the bearing, apparently dates back to 1886 in the United States when a patent was taken out for the idea.

A German investigator in 1917 advanced the claim that the spacing, number, and nature of depressions or recesses in interrupted surface finish are of major importance in minimizing friction, frictional heat, and wear, and increasing the load-carrying capacity of bearings.

The idea of filling such recesses in interrupted surfaces with some lubricating or other substance, to augment regular bearing lubrication, apparently originated in 1926. The use of other metals as filler materials, such as copper, silver, tin, or lead, to accomplish increased heat conductivity, anti-scuff and score resistance, etc., may be controversial, but apparently was sponsored in England by Shaw.

In the United States, at least one investigator had developed a grid bearing surface comprising the use of two different metals, as shown schematically in Fig. 2, by approximately the mid-thirties (Etchalls and Underwood, 1935). This was used for high loads for operation

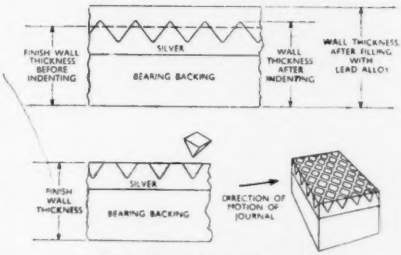
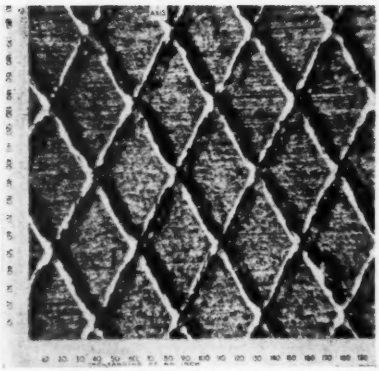


Fig. 2. Silver and Lead Grid Bearing
Bearing sections show indention detail.

with hydrodynamic full fluid film lubrication. Higher pressures in this type of bearing, particularly where alternating loads are imposed, require a larger number of small indentations. For conditions of moderate loads and boundary lubrication, the indentation representing about one-fourth to one-half the bearing area may not be filled.

Several years ago, an engine manufacturer, faced with increased speeds and loads beyond those originally designed for, and with consequent high oil-temperature, experimented with interrupted surfaces to secure longer life of crankshaft connecting rod bearings (Allen, 1941). An abraded surface was used which had a relatively smooth actual load-carrying surface of about 5 to 10 micro-inches, root-mean-square (0.000125 to 0.00025 mm.), peppered with deeper, elongated, irregularly



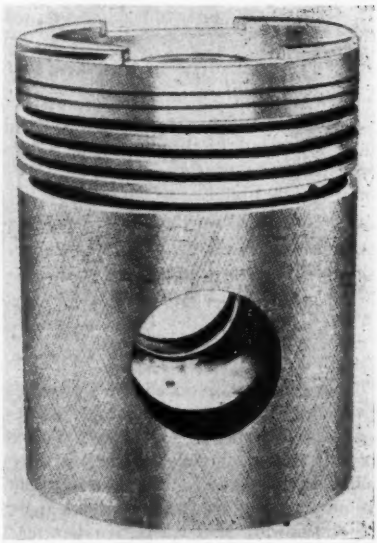
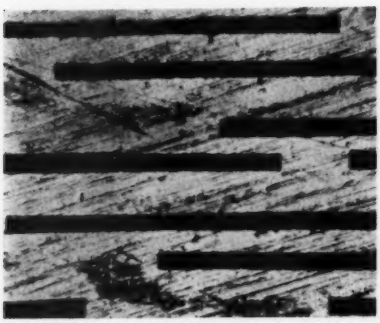


Fig. 3.—Knurled, Unfilled Surface used on a Cast-Aluminum Diesel Engine Piston.

spaced depressions to produce an overall roughness reading of 35 to 50 micro-inches. Despite very encouraging road test results, this project was reported ultimately abandoned because it was impossible to get dependable reproducibility in production machining.

A Diesel engine manufacturer, having an unexpectedly severe condition in a supercharged engine, is reported to have obtained complete relief of piston scuffing by use of the finish shown in Fig. 3. This is a plain, unfilled, knurled finish on aluminum.

An American aero-engine manufacturer, exploring the possibilities of using nitrided steel piston rings in SAE 4140 steel aircraft engine barrels, tried individually: knurling, "scarified" hone abrading, and masked chemical etching with definite indications in all tests of elimination of the "chatter" marks



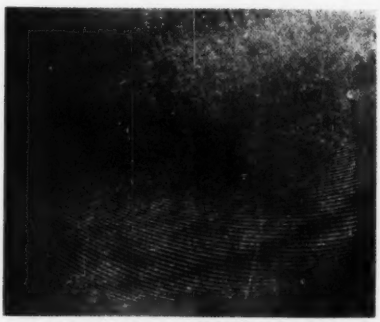


Fig. 4.—Knurled, Unfilled Surface Tested in an Aircraft Engine Cylinder Barrel.

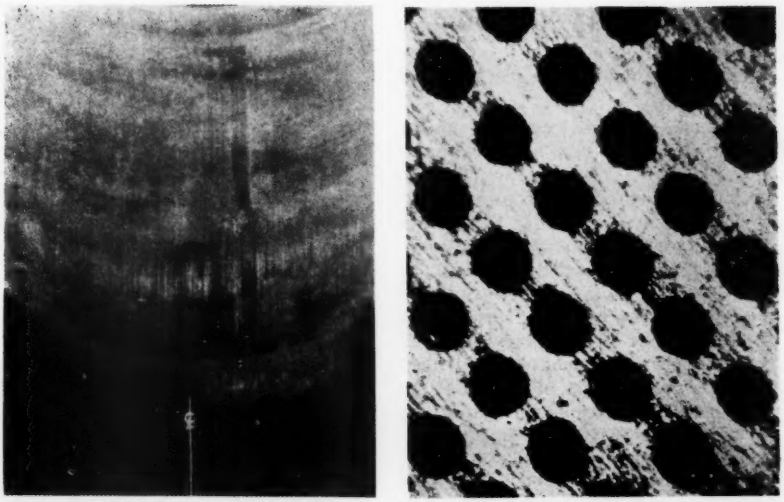


Fig. 5.—Masked, Chemically Etched Surface Tested in Aircraft Engine Cylinder Barrels.

which had resulted from using steel rings on smooth-honed steel barrels (Continental Aviation Report). The knurled finish used is shown in Fig. 4.

Smooth honing followed by masked chemical etching to generate voids only as shown in Fig. 5, was next tried, but was not comprehensively or further explored when it was found that filler material then available could not be

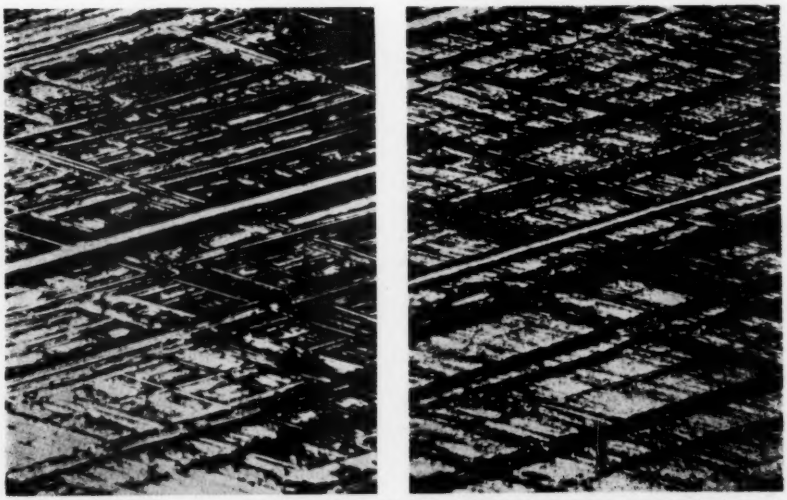


Fig. 6.—Microhoned, "Scarified" Surface Finish adopted in the Production of Aircraft Engine Cylinder Barrels.

dependably retained in the pits. Here shown are the 40-mesh and 65-mesh patterns tried, which had respectively 1600 and 4225 pits per square inch of surface area.

Next, a very coarse grit honing operation, followed by a very fine grit honing operation was tried, to produce the "scarified" surface finish shown on the right-hand side of Fig. 6. Wear after 50-hour endurance and 9¾-hour cruising tests was reported too slight to be measured. Lacquer formation was almost continuous over the surface, but the steel rings showed very high oil consumption. Subsequent tests with filled surfaces proved that filling gave the equivalent of the lacquer formed in long periods of running, providing protection against surface disturbance during running-in, and providing a continuous oil ring contact for better oil control. This scarified interrupted surface finish was ultimately adopted for production because tests showed freedom from surface disturbance of any sort, and this processing was readily available with standard equipment.

Comparative tests of various interrupted surfaces were run by another laboratory on a wear testing machine, which reciprocates a sample section of a cylinder bore against a stationary sample of a piston ring at 630 reciprocations per minute under no load for the first fifteen minutes, 600 lb. per sq. in. for the next fifteen minutes, and thereafter with increases of 200 lb. per sq. in. every five minutes until scuffing occurs. All tests here reported were run with cast-iron rings, ground, and under standard lubrication of 1 cu. cm. of oil per min.

The surface shown in Fig. 7, was honed to 4 micro-inches on a centrifugally cast graphitic steel and scuffed at 5200 lb. per sq. in. This same surface, when electro-etched with a cross-hatched pattern of intersecting lines at 45 deg. from the vertical, each 0.005 inch wide, 0.002 inch deep, and 0.050 inch apart, comprising about 18 per cent void area, scuffed at 10,000 lb. per sq. in., or about double the load-carrying capacity of the uninterrupted surface.

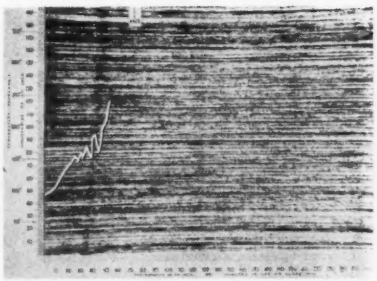


Fig. 7.—Scuff Record of Laboratory Cast Graphitic Steel Test Sample, Smooth-micro-honed to 4 Micro-inches (0-0001 mm.).

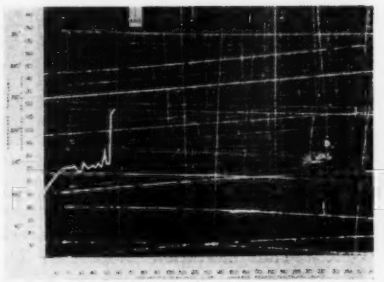


Fig. 8.—Scuff Record of Smooth-micro-honed Nitralloy Test Sample.

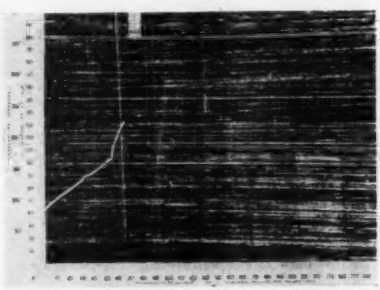


Fig. 9.—Scuff Record of S.A.E. 4140 Steel Laboratory Test Sample, Smooth-micro-honed to 4 Micro-inches (0-0001 mm.).

Examples of other material combinations include a smooth-honed Nitralloy

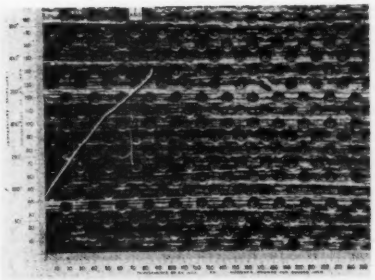


Fig. 10.—Same Sample as in Fig. 9, Masked and Chemically Etched to approximately 30 per cent Void Area, Shows 40 per cent Better Scuff Resistance.

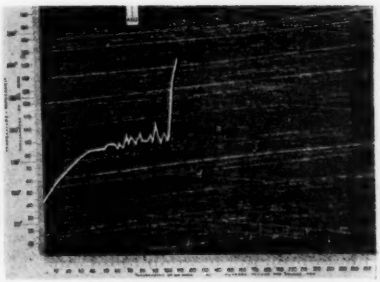


Fig. 11.—Scuff Record of 14-Micro-inch (0-0003 mm.) Microhoned Cast-Iron Laboratory Test Sample.

surface, which scuffed at 5600 lb. per sq. in., shown in Fig. 8, the SAE 4140 sample, Fig. 9, honed to 4 micro-inches, produced scuffing at 6600 lb. per sq. in., and the same SAE 4140 honed surface, masked and etched to produce pitted cells about 0.009 inch diameter, sixty-five to the lineal inch for an estimated 30 per cent void area, as shown in Fig. 10, scuffed at 9200 lb. per sq. in.

The next typical test sample, representing cast-iron cylinder bores, honed to 9 to 14 micro-inch finishes, serves as a baseline comparison with all similar test samples.

The 14-micro-inch honed surface, Fig. 11. scuffed at 11,000 lb. per sq. in.

The following surfaces as-honed to approximately 9 micro-inch were degreased and sprayed with a clear coating having a lacquer basecoat. Intersecting lines at 45 deg. to the axis were scribed through the coating. Surface material exposed through these lines was deplated in an ammonium chloride solution, with void area varying according to the width and distance apart of the lines.

The surface shown in Fig. 12, having 9 per cent void area, scuffed at 11,800 lb. per sq. in.

The surface with 27 per cent void area (Fig. 13), scuffed at 15,800 lb. per sq. in.

This exploration of various mechanical and other chemical and electrolytic

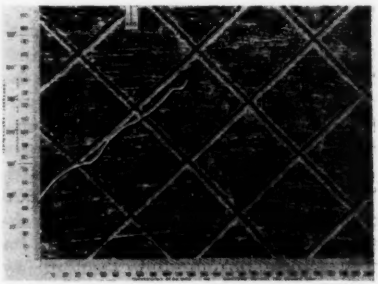


Fig. 12.—Scuff Record of Masked, Scribed and Chemically Etched Laboratory Test Sample, of approximately 9 per cent Void Area.

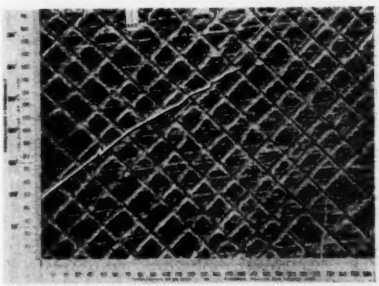


Fig. 13.—Scuff Record of Masked, Scribed and Chemically Etched Laboratory Test Sample, of approximately 27 per cent Void Area.

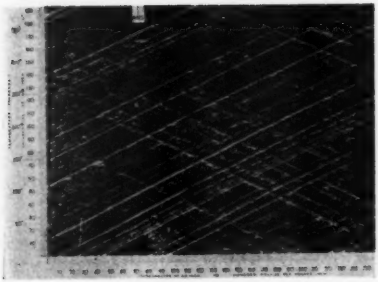


Fig. 14.—Hone-abraded, "Scarified" Surface Finish of Overall 80-120 Micro-inch (0-002 to 0-003 mm.), Test Sample Scuffs at 13,000 lb. per sq. in.

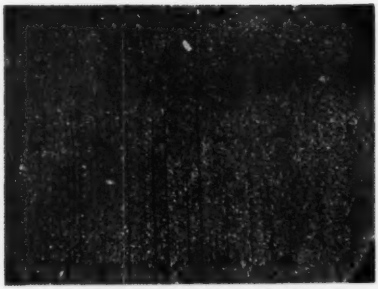


Fig. 15.—Knurled, Chemically Etched Cast-Iron Surface of 75 per cent Void Area Used in One Laboratory Test Engine Cylinder Bore.

generative methods reported in following sections has resulted in present concentration by this laboratory in a program of testing thoroughly all possible ramifications and possibilities of the scarified hone abraded finishes mentioned above, and as here shown typically in Fig. 14. This particular test surface, in the mid-range of roughnesses being studied, is relatively of 80 to 120 micro-inch overall roughness, with probably less than 10 micro-inch finish on the actual fine-abraded load-bearing plateau areas. It scuffed at 13,000 lb. per sq. in.

Another laboratory, using the single-cylinder test engine shown in Fig. 15, to determine pre-ignition ratings of aircraft engine sparking plugs, experienced severe difficulties with piston ring scuffing (Heron, 1946). Knurling was copied from others, but in this case the knurl marks are 0.004 to 0.006 inch deep, spaced 128 pits to the square inch of surface for an overall void area of only ¾ per cent, and with total volume of pits per square inch of 0.2 cu. mm. or about 1/100 of a drop. One of these bores, etched, and coated with manganese iron phosphate, and operated for

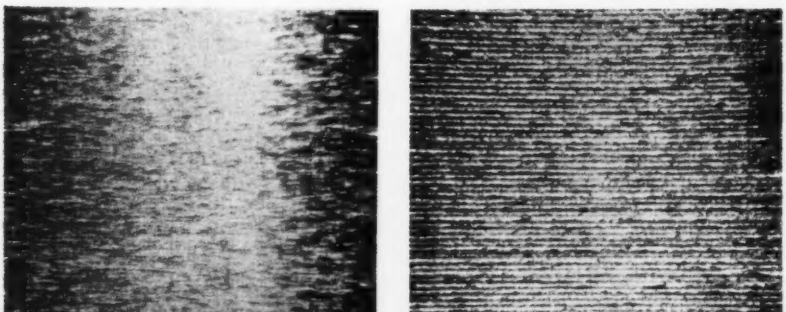
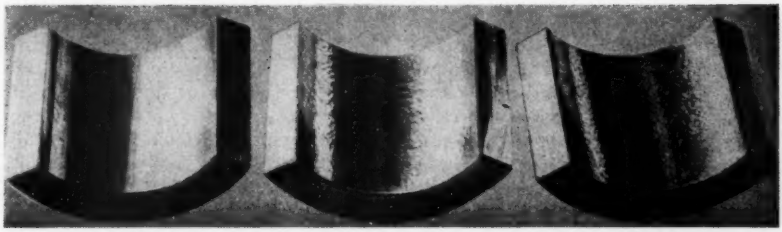


Fig. 16.—Combination of Fine-Ground Spindle and Rougher Scraped or Bored Bearing, Used by a Machine-tool Manufacturer.



(*a*) (*b*) (*c*)

Fig. 17.—Test Piece with Channel-type Interrupted Surface. (*a*) Smooth-honed. (*b*) Smooth-honed and mechanically peened. (*c*) Smooth-honed, mechanically peened, and smoothed-honed again to create plateau tops in one plane.

3700 hours with washed air, showed very low oil consumption and almost complete lack of scuffing; a maximum wear of 0.001 inch within the ring travel, and a *total absence of bell mouthing at the top of the bore,* even with ⅛ inch wide rings.

In recent years there has been a most interesting development in machine tool plain spindle bearings. In one particular use of plain bearings, shown in Fig. 16, it was formerly standard practice to grind the spindle to about 10 to 15 micro-inch finish, assemble it in a scraped bearing of approximately 35 to 40 micro-inch roughness, with clearance of 0.002 to 0.0025 inch. This spindle was operated with oil of about 75 to 100 viscosity, and ran at about 180 deg. F.

This bearing was ultimately refined and successfully used in production machines with a fine-ground spindle of 2 to 3 micro-inches in a scraped bearing of 35 to 40 micro-inches assembled with a clearance of about 0.00075 inch. It was operated with white kerosene of about 30 viscosity, and the bearing ran very cold; in fact, it felt clammily cold to the touch.

The running temperature of a bearing under speed and load is considered by some as probably a good indicator of performance. One investigator has commented: "Our experience with considerably higher speeds, of the order of 25,000 r.p.m. and above, have indicated very sharp rises in power consumption due to the high rate of shear of the oil film at such speeds. It seems to be pretty well established that the power requirements for an extremely high-speed spindle running in tilting pad bearings would be several times the power requirement needed for a spindle of the same speed running in ball bearings."

Many investigators believe that residual compressive stress is considered desirable in bearing surfaces. This result is obtained by honing, roll, and, or alternatively, ball peening. One such process, called "Micropeen," is shown in Fig. 17. The first process is to obtain a smooth-honed finish to generate an accurate control of size and geometry, then Micropeen to harden the material mechanically by cold working and at the same time generate a pattern of depressions for relief of the surface area. This is followed by finish honing to any desired operating surface finish on the plateaux which are the actual load-bearing areas. This process is considered economic, and controllable for production of accurate geometry, surface character, and size. It may be advantageous to eliminate the use of nonferrous metal bushings in such assemblies as the wrist pin bearing of automotive connecting rods, and other similar parts.

CHEMICALLY ETCHED INTERRUPTED SURFACES

The production of microscopic recesses in load-bearing surfaces by chemical action was first advocated (so far as could be determined) by Loudenbeck in 1915. The process comprised the production of a self-lubricating bearing by *first* "smoothing the surface, then etching it to produce recesses, and finally *filling* the etched surface with graphite particles to thereby reproduce the original smooth bearing surface." Later inventors have followed with mask etching to produce a specific pattern and disposition of recesses, either on a previously finished surface, or one which may require subsequent finishing of the load-bearing areas; the adaptation of the half-tone photolithographic process for improved pattern etching; chemical treatments after a recessing operation, and others.

Since final surface finishes generated by overall chemical or anodic bath etching usually generate some chemical reaction or deposition of oxides, sulphides, and phosphates upon the base metal, these processes have been widely termed as surface coating processes. In general, the desirable properties of satisfactory coatings are said to be *(Mechanical Engineering)* that:—

1. It must respond to varying degrees of pressure.
2. It should have mild abrasive action to aid the running-in of the two parts.
3. It should slightly contaminate the metal surface to prevent welding.
4. It should absorb oil and thereby improve lubrication.
5. It should etch the surface, leaving a somewhat porous surface to provide an escape for worn-out particles and to form minute reservoirs for oil.

The automotive industry today uses three main types of chemically etched surface coatings for load-bearing areas of various parts. These processes have been fully described in the available literature. It is interesting to compare the user's own evaluations in each case of the attainment of the above objectives.

Caustic sulphur etching is commonly used on cast-iron cylinders, pistons, piston rings, and on steel gears, camshafts, tappets, and similar parts. One of the major users of this treatment claims (Jackson, 1941) that it:—

1. Removes undesirable components of surface composition such as free ferrite, and strain-hardened material resulting from previous processing.
2. Deposits certain chemical end-products on the surface, in this case ferrous oxide and ferrous sulphide in a porous, matt surface.
3. Changes the surface configuration by forming pits which are partially filled with porous sulphide-oxide contents which ensure lubrication during at least the critical period of boundary lubrication during "run-in."

Though limited to chilled or cast iron parts, the ferrous oxide treatments are credited with excellent results on cast iron piston rings (Teetor, 1940), and on chilled iron cam followers and tappets *(Mechanical Engineering;* Young and Davis, 1947). This finish is shown in the photo-micrograph in Fig. 18, and

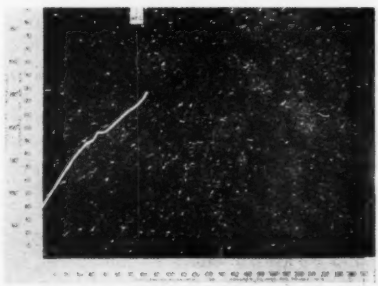


Fig. 18.—Photomicrograph and Scuff Record of Caustic Sulphur Etched Cast-Iron Laboratory Test Sample, 28 Micro-Inches (0·0007 mm.).

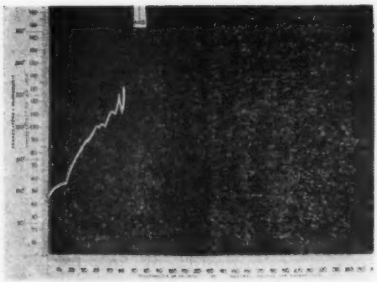


Fig. 19.—Photomicrograph and Scuff Record of Manganese Iron Phosphate Coating on S.A.E. 4620 Steel Test Sample, 15 Micro-inches (0-004 mm.).

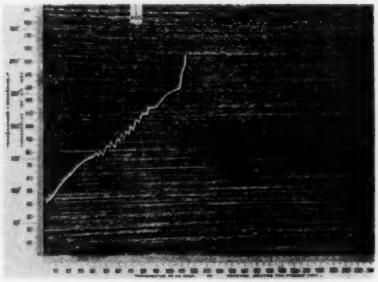


Fig. 20.—Photomicrograph and Scuff Record of Manganese Iron Phosphate Coating on S.A.E. 4620 Steel Laboratory Test Sample, 73 Micro-inches (0-0018 mm.). Shows 80 per cent Increase in Scuff Resistance.

as here shown has a roughness of 28 micro-inches (0.0007 mm.).

Manganese iron phosphate coatings are being adopted more frequently for cast-iron bearings, cylinder bores, pistons, piston rings, chilled iron cam followers, steel gears, universal joint trunnions, valve stems, pump shafts and other parts. The advantages claimed for this process are that (Young and Davis, 1947; Darsey, 1941; Technical Bulletin, Parker Rust Proof Company):—

1. It deposits a porous deposit of manganese and iron phosphate on the surface, which prevents welding.
2. This coating is mildly abrasive to accelerate the "running-in" of the parts.
3. The etching action of phosphoric acid also produces a porous surface under the coating which provides an escape for debris and furnishes minute oil reservoirs that remain even when all traces of coating have disappeared."

The following typical wear machine tests were run on this type of manganese iron phosphate coated surface by a previously mentioned laboratory. The cylinder material was SAE 4620 steel, carburized and ground to various roughnesses, as noted. Such a surface, Fig. 19, ground to 7 micro-inches before etching, and etched to 15 micro-inches, scuffed at 6000 lb. per sq. in.

In Fig. 20, the sample was ground to 65 micro-inches, finally etched to 73 micro-inches, and scuffed at 11,400 lb. per sq. in.

Apparently, in most of the applications which have been reported, the thickness of this coating is usually not more than 0.0002 to 0.0003 inch, but can be applied up to 0.001 inch thick. Most users apparently believe that the thinner range of coatings have greatest value as a rule during the running-in process, offering protection against metal-to-metal contact until chemical-physical stability has been established.

ELECTROLYTICALLY GENERATED INTERRUPTED SURFACES

The production of interrupted type surface finishes by conventional anodic etching has also been investigated experimentally. Such a typical surface may be, for example, anodized in a 1 per cent hydrochloric acid solution, with current density of 0.025 amp. per sq. in., in a room temperature bath. This application oxidizes the free ferrite, which can be brushed away leaving the steadite

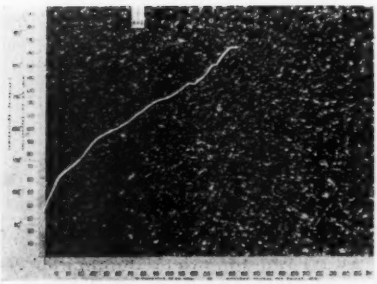


Fig. 21.—Photomicrograph and Scuff Record of Anodically Etched Cast-Iron Surface, 50 Micro-inches (0-0012 mm.).

network projecting to form an interrupted surface of irregular projections. It is applicable only to cast iron.

The surface in Fig. 21, anodized 15 minutes to produce a 50 micro-inch surface, scuffed at 15,600 lb. per sq. in.

While the idea of electro-deposition of a wear resistant coating of metal on bearing surfaces dates back at least to 1930, control of the plating process to produce pits, depressions, or crevices by anodic etching in hard chromium plating as shown in Fig. 22, was first mentioned in the process which has become known under the term "porous" chromium plate (Van der Horst, 1940, 1941; Pyles, 1944). This development was first introduced commercially in the marine and stationary Diesel-engine field; fortunately it was available during the 1939-45 war for salvage of many thousands of aircraft and tank engines

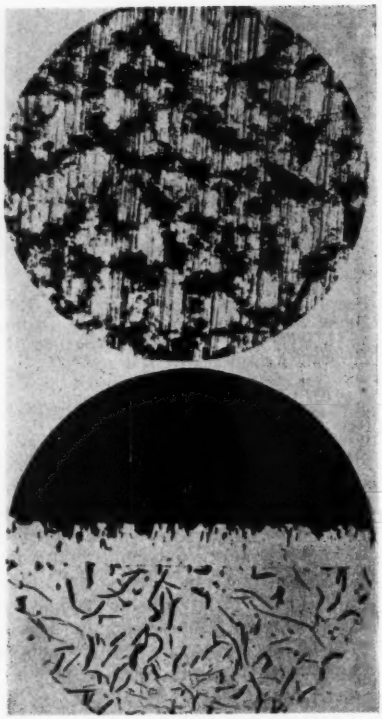


Fig. 22.—Photomicrographs of "Porous" Chromium-plated Interrupted Surface Finish, with Pitted and Some Porous-type Interruptions, known as "Normal Type" Porosity.

TABLE 1. NORMAL TYPE POROSITY

Porosity range, per cent	Results
2 to 10.....	A definite high rate of wear and a high percentage of scuffing.
10 to 20.....	Marginal. A tendency towards a high rate of wear and minor scuffing.
20 to 35.....	Fig. 23. These cylinders gave the least scuffing and the lowest rate of wear.
35 to 50.....	A tendency towards high wear.
50 to 60.....	High piston ring wear, heavy scratching, some scuffing.
Above 60.....	Extreme wear, scuffing.

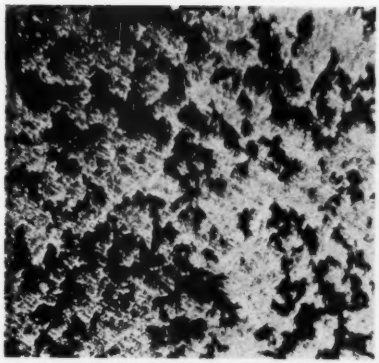


Fig. 23.—Photomicrograph of 20 per cent to 35 per cent Void Area Normal Type Porous Chromium-plated Finish. Produced the least scuffing and lowest rate of wear in military service.

in those theatres of operation where sand and grit caused steel cylinder barrels to wear completely out in as low as 30 hours of operation. The story of this development has now been released by official sources (Shaw, 947; Schmiedlin, Miller, and Esterly, 1944; Bobrowsky, Machlin, and Onnstein, 1944; Schmiedlin and Ault, 1945; Johnson and Swikert, 1946; Johnson and Anderson, 1946).

In this first step in the development, the surface is comprised of irregularly disposed and isolated relatively shallow pits, depressions, or crevices surrounded by connected plateau areas; this is known as normal type porosity. The actual load-carrying plateau areas are estimated to be about 5 to 7 micro-inches (0.000125 to 0.000175 mm.), while an actual profilometer measurement of overall roughness might be anywhere from 50 to 200 micro-inches. Aircraft engine service tests on this type of finish showed the results in Table 1.

A modification of this development, involving a time differential, different bath constituents, current densities and temperatures in the plating process, produces the interrupted surface shown in Fig. 24. This is a surface consisting

Fig. 24.—Photomicrograph of Channel-type Porous Chromium-plated Interrupted Surface Finish.

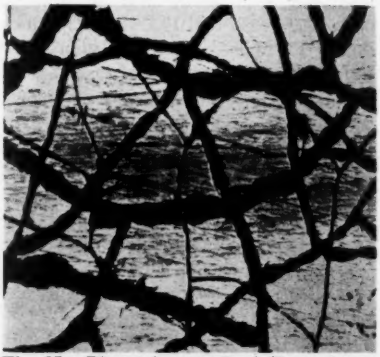


Fig. 25.—Photomicrograph of Channel-type Porous Chromium Surface. Has plateau sizes from 0-010 to 0-020 inch (0-254 to 0-508 mm.) which produced least scuffing and lowest wear.

TABLE 2. CHANNEL TYPE POROSITY

Plateau size, inches	Results
0.002 to 0.005...	Almost certain scuffing, breakdown of chromium surface.
0.005 to 0.010...	Occasional scuffing.
0.010 to 0.020...	Good scuffing resistance and best wear characteristics. Fig. 25.
0.020 to 0.030...	Good scuffing resistance and good wear characteristics.
0.030 to 0.040...	Occasional scuffing.
0.040 to 0.060...	Almost certain scuffing and poor wear characteristics.

of isolated plateaux surrounded by relatively deeper connected channels; and is thereby named channel type porosity. Aircraft training service tests have shown no measurable wear on this finish after 500 hours of flight operation. Experiments established the following data for this type surface interruption, as shown in Table 2.

Channel widths apparently are not critical, ranging from 0.001 to 0.003 inch, with an average of 0.002 inch in depth, but it was found highly essential that all debris from processing should be removed by high pressure blast of a cleaning fluid directed normal to the cylinder surface. The objective of production practice for salvage processing of aircraft engine barrels by this method was therefore established at plateau sizes between 0.010 and 0.030 inch honed to approximately 1 micro-inch finish. Several hundred thousand military aircraft engine cylinders so processed have given entirely satisfactory results.

Further tests indicated the value of an extremely small plateau size for best possible wear rate and scuffing resistance, but small plateau peaks less than 0.010 inch diameter break off easily, owing to the low tensile strength of chromium. Seeking some such type of surface interruption which would eliminate engine initial running-in time after cylinder replacement and permit starting and accelerating to take-off power in 60 seconds, and running at take-off power from 20 to 60 minutes, a third type of interrupted surface finish, named deep-smooth has been developed. This is shown in Fig. 26, and consists of an infinite number of small peaks. It is prepared by first lapping with a dummy piston and rings and with a mixture of several grades of a commercial oil mixed lapping compound, which produces an average 43 micro-inch finish.

Fig. 26. — Photomicrograph of "Deep-Smooth" Porous Chromium Surface of 24 Micro-inches (0-0006 mm.).

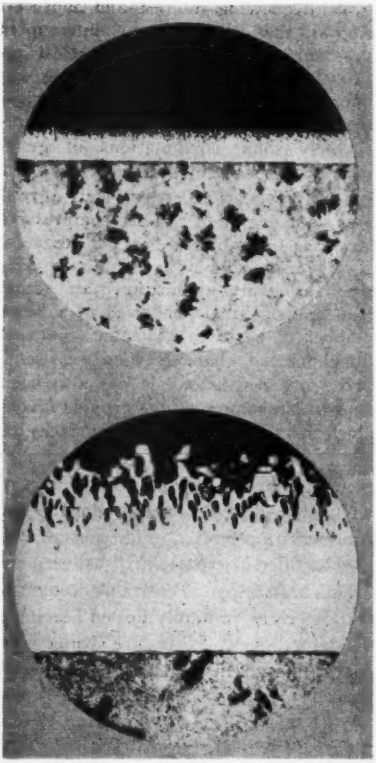
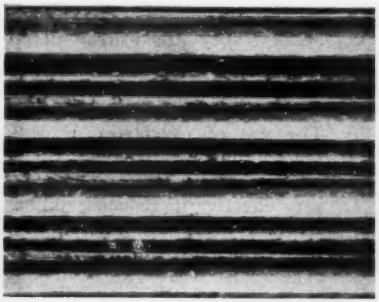


Fig. 27.—Photomicrograph of Spongy Porous Chromium-plated Surface used in production by a Piston Ring Manufacturer.

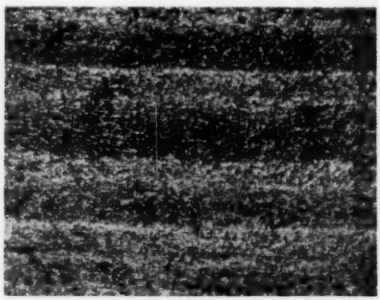
After cleaning, the surface is again lapped with a dummy piston and rings, with extra fine pumice mixed with paraffin oil to a roughness reading of 24 micro-inches. This surface has operated at take-off power for one hour with practically no observable difference in the finish. Though not at present completed, the testing of this third type of interrupted chromium-plated surface points to its adoption in the very near future for overhaul processing of military aircraft engine barrels.

Extensive work has been done to determine the operating characteristics required for use of porous chrome-plated piston rings. One manufacturer has established certain general conditions, as follows: a porous chromium ring to be used in the top groove on each piston, having nearly the entire face surface relieved as shown in Fig. 27, a micro-inch reading of 30 to 90, and possessing "chalk," as shown, promotes rapid seating on all rings where the cylinder bore surface is honed to 15 to 25 micro-inches of roughness on a 30 deg. helix cross-hatch. It is further considered beneficial in carrying oil during the critical running-in-period.

An early war-time investigation established the necessity for smooth-honed plateau surfaces when it was discovered desirable to run-in (green run) aircraft engine cylinder barrels, with surface



× 200 dia.
Fig. 28.—Surface Condition of Aluminum, Diamond-turned.



× 200 dia.
Fig. 29.—Surface Condition of Aluminum after Tin-plating.

finish of 45 per cent to 60 per cent porosity with one set of piston rings, and then replace them with new rings for engine installation (Schmiedlin, Miller, and Esterly, 1944; Bobrowsky, Machlin, and Onnstein, 1944). Tests to determine methods for improving uniformity of pore sizes and locations on chrome plate were tried (Schmiedlin and Ault, 1945), but apparently were inconclusive because time has not yet permitted comprehensive program testing. The use of fillers or coating materials on channel type chrome-plated cylinders was investigated (Johnson and Swikert, 1946) with the result that a silver overplate was adjudged better than either a lead overplate or a colloidal graphite base paint for reducing specific oil consumption, for increased lacquering of the surface, and for some reduction in ring wear. However, initial oil consumption was most practically and positively reduced by the use of 1° deg. taper-face compression rings in the top three piston grooves, installed to scrape down, and reduced at least one-third through the use of choke bores (Johnson and Anderson, 1946).

With growing interest and appreciation of the possibilities of interrupted type surface finishes as a practical approach to increased load-carrying capacity, longer life and better lubrication of bearings, it is evident that ultimately some combinations of generative methods will become defined for various applications. It is widely agreed that some degree of residual compressive stress is desirable in bearing surfaces. It has been shown in the above data that for many applications an actual load-bearing area comprised of a large number of peaks or plateaux separated by connected channels increases scuff and score resistance. It has also been shown that some coatings provide favorable chemical deposits on surfaces in addition to a substantial amount of surface interruption; and that they improve wettability and lubrication during the running-in period. Solid non-ferrous metal fillers, particularly silver, is demonstrated as a desirable constituent, especially in highly loaded bearings. Therefore, new ideas are being encouraged and new methods are being tried.

CONCLUSIONS

It is evident that a great deal more development and experimental work must be done in a wide number of applications before any very definite conclusions can be drawn. Apparently those which may be indicated now include:—

For boundary lubrication:—

1. For moderate loads voids are preferably not filled but total surface area may benefit with a chemical etch.
2. Plateau load-carrying areas should be relatively smooth, from 1 to 10 micro-inches (0.000 025 to 0.00025 mm.), though overall surface roughness may be from 50 to 200 micro-inches.
3. Area of individual plateaux is critical as to size and disposition with increasing load.
4. Connected channels are desirable because they are usually deeper and consequently are critical as to cleaning of process and wear debris before the running-in period.
5. The use of overplates or solid surface films on boundary lubricated surfaces have been found very effective in reducing sliding, friction, wear, etc., under conditions which are found in engine cylinder barrels.

For hydrodynamic lubrication:—

1. For high unit loads, particularly alternating type, the voids should

be small in size and filled or overplated with a non-ferrous filler to accelerate compatibility of bearing couple materials and the lubricant.

2. Solid fillers or overplates are desirable to provide "embedability," elasticity, conformability, thermal conductivity.
3. For high speeds (25,000 r.p.m. and over) and heavy loads, any type of plain bearing, relieved surface or not, cannot be expected to replace anti-friction (ball or roller) type bearing; this is because of the inherent characteristic of lubricants to generate frictional heat by shear effect within the oil film itself.

For all bearing surfaces:—

A final chemical etch on a bearing surface seems desirable to provide a mild abrasive action to aid running-in, to contaminate the surface against welding, to provide a porous or chalky surface structure for trapped oil, and as an escape for dirt or debris particles, particularly where close clearances are required.

ACKNOWLEDGMENTS

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APPENDIX

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SAND AND DUST EROSION IN AIRCRAFT GAS TURBINES

ACKNOWLEDGMENT

This article, reprinted from the December 1949 issue of *Aero Digest*, is by MR. J. E. DEREMER, of the Power Plant Laboratory, Air Material Command. The article is condensed from a paper presented at the S.A.E. National Aeronautics Meeting in Los Angeles, October 1949.

Although the success of methods developed to meet the problems of desert operation is established for reciprocating engines, the gas-turbine powerplant presents new problems. The memory of the frantic, expensive, and necessarily rapid development of carburetor air filters during the war years is still fresh. The desire to avoid an occurrence of this sort of accelerated development with gas turbines prompted the USAF to investigate the vulnerability of this type of powerplant to desert operations.

The basic questions to be answered are whether there are *any* effects due to operation under sand and dust conditions that will result in operational hazards or require frequent engine changes, or, if there are no rapid gross effects, what kind of life and performance can be expected from gas turbines operated under desert conditions?

Due to the differences between the gas-turbine and reciprocating engines, the great amount of data concerning reciprocating-engine life and the various effects of dust on that type of powerplant gives a few clues as to what may be expected of gas turbines operating under dust conditions. The greater clearances of primary components in the turbines would appear advantageous. The higher rotational speeds and dependency of thrust on maintenance of certain dimensions would appear disadvantageous. Air-supply tubes to engine-mounted accessories such as generators, alternators, cabin-conditioning equipment and air turbines provide the means of dust entrance to this equipment, with possibly destructive effects.

The compressor would seem to be the most vulnerable component of the turbine engine, particularly the axial-flow type with its thin blades. The burners and fuel nozzles should not present much difficulty due to the higher-strength materials, large dimensional tolerances and the improbability of nozzle clogging. The nozzle diaphragm and turbine wheel, of higher-strength materials—thus less subject to erosion than the compressors—have critical surface contours and are more subject to failure, due to high operating temperatures.

The early-type bearing-lubrication system utilizing compressor air bleed for oil-mist lubrication presents possibilities of rapid and serious effects in

the dust clogging of the air filters used in the system. Current integral-lubrication systems avoid this possibility. However, all compressor air-bleed lines are subject to sand collection and would impair services such as thrust balancing and cooling.

The general effect to expect in current gas turbines under dust conditions would seem to be a gradual deterioration of performance rather than extensive and rapid failure of components with the attendant hazards. Possible exceptions to this would be earlier failure in rotational parts, induced by dust erosion or compressor damage from poor airflow conditions occurring in an eroded duct system. Periodic checks on filters in compressor air-bleed lines would avoid the hazard of clogged balance or cooling air services.

The problem would appear to be one of measuring time effects in the deterioration of performance of the basic engine and securing data to be measured. Data on engine accessories must be obtained to determine their limits and filtration requirements.

The relatively limited operations of gas turbine-powered aircraft (particularly axial-flow compressors) has prevented any organized service testing of aircraft under desert conditions. Present tactical and operational procedures with gas turbines hinder an accurate evaluation of the severity of dust operations. A controlled test on a single aircraft, imposing dust conditions chosen from experience in the operation of reciprocating-engine aircraft, appears to be the only means of obtaining an idea of the problem.

The first test of this type was conducted at the Eglin Field Climatic Test Hangar during November, 1948, on a J47-GE-1 axial-flow turbojet installed in an F-86 airplane. The general test set-up consisted of a sand hopper and an injection tube into the inlet, and an air hose connected with it to blow the sand into the inlet.

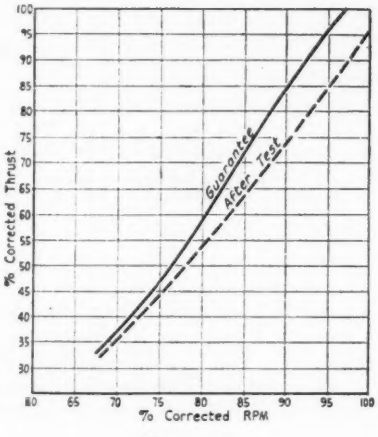

Figure 1

The sand concentration injected into the engine was 1.36 lb/sec at maximum rated power. This can be considered an accelerated test, in that the concentration of sand and dust was some 29 times the amount that would be ingested at the same power with dust conditions at the inlet as called for in Specification AN-F-30a.

The test was terminated at the end of a period of three minutes, under these

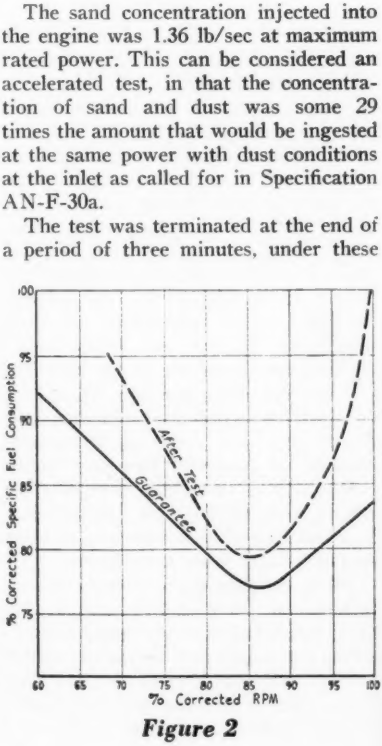

Figure 2

conditions, when a circumferential rupture 3-in. long and ¼-in. wide occurred in the exhaust cone 3-in. aft of the turbine flange, probably induced by clogged burners. A total of 343.8 lbs. of sand had passed through the engine.

A calibration of the engine, following replacement of the tail cone and mechanical inspection only, yielded the performance illustrated in Figs. 1 and 2, as compared with guarantee ratings for the J47-GE-1.

BREAKDOWN FOR STUDY

Following the calibration, a complete tear-down was made and the condition of the parts was noted. The starter-generator had severe commutator pitting and heavily-scratched contact surfaces of the brushes. The rear tips of 23 guide vanes caused V-type vibration marks on the gearcase-support segment of the forward frame.

The compressor-rotor blades were eroded severely at the blade tips and trailing edges. Erosion was moderate on all blade leading edges. The first-stage blades were bent at the trailing edges. All the anodizing surface treatment was eroded from aluminum spacers and wheel rims, as well as the dichromate surface treatment of the steel-wheel rims. The compressor-stator blades were eroded severely at the trailing edges near the base in the forward stages.

Erosion was more severe and localized closer to the blade tips in the later stages. The compressor casings were eroded enough to indicate the pattern of airflow in the forward stages. The extraction air holes were eroded, giving the circular holes an elliptical shape with the long axis parallel with the airflow. All the dichromate treatment was eroded from the magnesium casing and stator rings and from the magnesium midframe in the diffuser section.

The inner liners of the combustion chambers were in excellent condition. Several outer liners were discolored, indicating hot spots during the sand-test operation. The bellows in each outer chamber was full of sand, prohibiting free expansion and contraction of the chamber. The drain line of No. 3 and No. 6 combustion chambers was broken. The transition pieces indicated slight and uniform erosion.

All nozzle-diaphragm blades indicated slight erosion at the blade trailing edge, with excessive erosion of the outer-blade trailing edge behind combustion chambers 5, 6, and 7. The turbine buckets had uniform erosion of all blade tips inward for ¾-in. on the convex side. *Zyglow* inspection of the wheel indicated no cracked buckets. The turbine shroud ring had three areas, 5-in. long and ¼-in. wide, burned away behind burners 5, 6, 7, and 8. The exhaust cone had a circumferential burned hole, 3-in. long and ¼-in. wide, 3-in. aft of the turbine flange behind burners 6 and 7.

A thorough inspection of oil lines, pumps, oil filters and bearings did not show any presence of sand. All balance air lines and air-cooling lines were packed with sand, permitting little or no effective airflow. The 1st-stage and 12th-stage compressor air seals were sand-blasted, but indicated only normal wear on rubbing surfaces. The sand that had collected inside the areas protected by the seals was in a fine powder form. The mid-frame and turbine-air oil seals indicated only slight blast effects. Static and dynamic balance of the compressor was unchanged by the dust erosion.

THE QUESTION OF SPEEDING UP

The rapidity of failure of the J-47 engine during the Eglin Field tests brought up serious questions regarding the validity of an accelerated test. As a result, Air Material Command set up a similar test on a YF-84 airplane, using

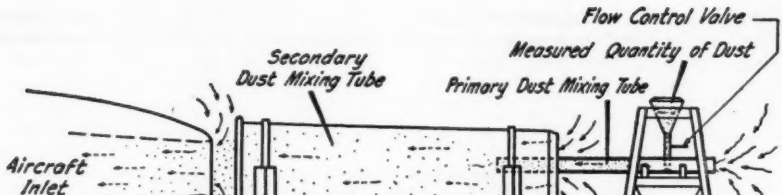

Figure 3

a J-35-A-15 engine, with the objective of duplicating actual operational conditions as closely as possible. After experimenting with several dust-injection methods, the arrangement illustrated in Fig. 3 was accepted.

The large mixing tube is an F-80 exhaust tailpipe with the larger opening facing the aircraft inlet. The smaller mixing tube is a 3-in. OD pipe. The low-pressure region at the aircraft air inlet produced airflow through both mixing tubes, the velocity dependent on the distance of the large tube from the aircraft air entrance. Turbulence produced in the large tube thoroughly distributed the dust over the entire aircraft air entrance, thus providing natural conditions without the need of auxiliary-power equipment.

The general test procedure used was to operate the engine to a power schedule that is followed from start to takeoff and climb, and from approach to landing and shut down. The dust weight-flow per unit of time that would be ingested at each of the power conditions was calculated, using the corresponding engine mass airflow at sea-level conditions. An average time that an aircraft normally remains at each of the power conditions was arrived at by timing many flights from Wright Field.

With the time periods for each power condition calculated, the total weight of sand that the engine would ingest for that period was obtained. The conditions used in each run were as follows:

Engine Condition	*Time*	*Total Sand*
Idle	1 minute	336 grams
Taxi—60% rpm	4 minutes	3240 grams
Takeoff—100% rpm	1 minute	1620 grams
Approach—Land—Taxi 60% rpm	4 minutes	3240 grams

The sand and dust were introduced after each power condition was reached and stabilized. This schedule was to be followed until engine failure requiring removal occurred, or a sufficient number of takeoffs and landings were accomplished to correspond with the normal overhaul period of the engine. For the F-84 aircraft, assuming fuel endurance of three hours per flight, and a 150-hour engine overhaul, this would amount to 50 takeoffs and landings. Periodic inspections were made of the progress of erosion by removal of the engine from the aircraft and partial disassembly of the engine.

REGULAR EQUIPMENT

Only the standard cockpit instruments were used for engine r.p.m., tailpipe temperature and fuel and oil pressures. In any further testing, a bearing air-pressure line will be instrumented to indicate the extent of clogging of the air filter in the line, to avoid engine-bearing failures.

The engine used was subjected to standard calibration prior to the dust tests to provide a basis for measurement of performance deterioration.

The schedule was followed for four runs at which time the engine was removed, partially disassembled and inspected. All internal surfaces exposed to airflow were quite clean, including the spark-plug electrodes—giving rise to possible use of small quantities (1 or 2 lbs.) of dust for periodic cleaning of the engine. Very minor erosion was observed and the engine was reinstalled and the schedule continued with visual inspection only until 15 runs were completed.

A hydraulic leak occurred, necessitating engine removal at this point and again the engine was partially disassembled and inspected. The progress of erosion was observed but no serious wear was apparent. The engine was reinstalled and the schedule was continued until 26 runs were completed. At this point, bearing failure occurred and the engine was removed, completely disassembled and inspected.

The bearing air filters were clogged severely with sand and dust. (Fig. 4) The cause of the bearing failure was not clearly the result of the clogged filters, in that *all* filters were clogged and the remaining bearings gave no indication of impending failure. Airflow checks on the clogged filters indicated that a considerable flow of air was obtainable still. However, No. 2 bearing was the most critical and the reduced airflow probably accelerated failure.

Erosion was apparent in all stages of the compressor and on the leading edges of the midframe islands. The starter-generator unit contained a considerable quantity of sand and dust, and the commutator and brushes were eroded and pitted. The compressor lands showed some pitting on the top half. Certain portions of the aircraft duct walls showed mild erosion: all other engine components showed little or no erosion.

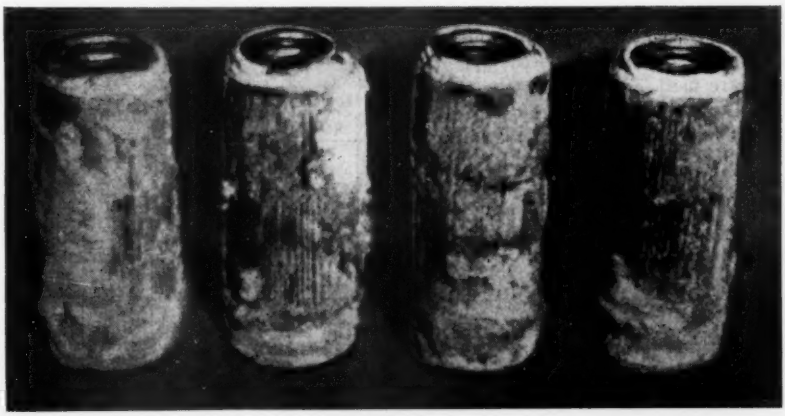

FIGURE 4.

The static and dynamic balance of the compressor and motor was not seriously affected.

Vibration readings during the calibration run after the test improved over the first calibration of the engine. Characteristic deposits of the fine dust were found on all surfaces, including compressor rotor and turbine blading, indicative of airflow conditions at those surfaces. This suggests an experimental technique similar to the use of strains for qualitative checks or internal-airflow conditions in a turbine.

At the completion of the 26 runs, a total of approximately 474.5 lbs. of sand and dust had been passed through the engine. On the basis of 3 hours per flight, this would be equivalent to 78 hours engine time. The effect upon thrust and specific fuel consumption indicated by a calibration run is illustrated in Figs. 5 and 6 as compared with the factory calibration run and a calibration prior to the dust test. Continuation of the J35 tests by AMC is planned in an effort to determine extreme limits of operation under sand and dust conditions.

Comparisons of results indicate that an accelerated test, using high sand concentrations in a short period of time, is not a valid test, in that erosion is much more rapid and severe for equivalent total quantities of sand and dust than when conditions approaching natural are used.

The conditions of the test were conservative in that the dust conditions existed for every simulated takeoff and landing of the aircraft during the total test, which may not be the case in actual operations; the sand and dust had a uniformly high abrasive quality; natural conditions will vary considerably; a separation effect may be exercised on the larger, more-abrasive particles by airframe air inlets of another type than the test inlet; sand and dust concentrations will be lower where engines are mounted high in the nacelles or wings.

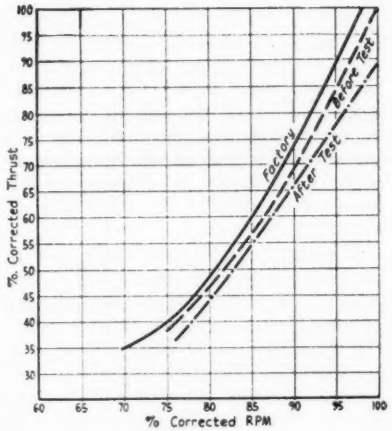

Figure 5

On the other hand, the conditions of the test were not conservative in that ground engine operation periods in sand and dust may be considerably longer under certain tactical conditions than those of the test; flight time and sand dust may be considerably longer in regions of frequent dust storms where concentrations of dust reach altitudes

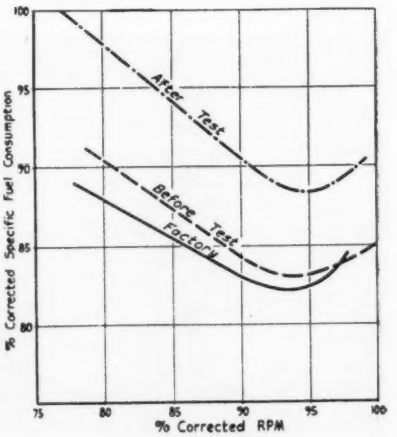

Figure 6

up to 8000 and 10,000 ft.: the force exerted on the ground by jet wakes in mass operations may produce concentrations of sand and dust greater than that used in the test which was based on measurement behind propellers.

ALLOWANCES

Results must be considered in relation to engine time in flight, compared with that on the ground, varying considerably between types of aircraft missions.

The data indicate that while aircraft gas turbines may not be rapidly or grossly affected they are not immune to sand and dust problems and present a situation worthy of further consideration. Any protective measure attempting to remove sand and dust from the intake air appears hopeless at this point because turbine performance is so dependent on maximum inlet duct efficiency. The most feasible approach appears to be the use of materials of high erosion-resistant qualities in the critical areas of the engine such as the compressor. In the meantime, consideration by the aircraft designer toward the location of duct entrances in lower dust-concentration areas or the use of flush-type entrances would alleviate the problem somewhat.

The comparative vulnerability of the centrifugal-flow compressor type turbine to the axial-flow compressor type turbine is an interesting problem. A test similar to that conducted on the J-35 engine is scheduled for the J-33 engine. Tests to evaluate the effect of sand and dust on auxiliary services from the compressor, such as cabin-conditioning and anti-icing systems, are planned.

VIBRATION DIAGNOSIS IN MARINE GEARED TURBINES

ACKNOWLEDGMENT

This article is by MR. H. G. YATES, member of the North East Coast Institution of Engineers and Shipbuilders. It appeared in its entirety in the Transactions, Volume 65, of that organization. Space limitations have prevented reprinting Appendices I-IV, The Notes on Apparatus, the discussions, both written and oral, and the author's reply. All appear in the Transactions. The paper deals mainly with vibrations of the forced resonance type induced by running machinery.

INTRODUCTION

There have been so many excellent papers on vibration read before this Institution and recorded in its *Transactions* that it is with some hesitation that a further paper dealing with the same field is now presented. The Author's justification is that in this paper the subject is approached from a somewhat different viewpoint. The study of vibrations in engineering is a relatively recent development by comparison with the long history of mechanical and marine engineering, mainly because vibration problems generally present themselves in a serious aspect only when dealing with machinery which runs at a high speed or in which the component parts are heavily loaded. It is therefore natural that in the early stages of development of any form of prime mover or power transmission, problems of vibration usually arise at a relatively late stage in the history of this development and most of such problems have been first experienced in practice and subsequently handled on theoretical lines.

Earlier work on vibration has dealt mainly with methods of predicting machinery characteristics in advance of construction or with methods of measuring and recording vibrations (in which is included noise) in running machinery. This paper describes a technique of investigation which has been built up over a period of years and which has for its object the determination of the source of vibration in any particular case with the primary aim of eliminating it. To the designer and the experimenter vibration phenomena may have considerable interest in themselves, but they only excite the attention of the marine engineer when they manifest themselves in an objectionable form and his sole concern is to get rid of these effects.

Vibrations in machinery may be divided broadly into two classes, namely, self-maintained vibrations and forced vibrations. The first class includes such cases as flutter of propeller blades, Aeolian tones and relaxation oscillations in which a system progresses more or less steadily in one direction until its motion is abruptly interrupted to be recommenced subsequently. In many such cases the distinction between the two classes has a borderline character. In engineering, self-maintained vibrations cover a very small proportion of the total.

The second class, which includes al-

most the whole of vibrations actually experienced in engineering, arises when a mechanical, acoustical or electrical system receives periodic disturbances from an outside source. In the majority of text books a distinction is made between forced and resonant vibrations but this distinction is one of degree rather than of kind. Almost every system has one or more natural resonant frequencies and its behavior under forced excitation is determined partly by the characteristics of the forcing and partly by the closeness of approach of the forcing frequencies to one of the natural frequencies. Thus the existence of such natural frequencies has always some effect on the response of the system even though this effect may be small if the forcing frequency is far removed from the frequency of excitation.

In vibration diagnosis it is frequently necessary to make rapid if approximate calculations of the behavior of a system subjected to periodic disturbances, and at such times it is convenient to express the mathematical equations governing such a system in a form which assists rapid calculation of essential parameters.

While every mechanical system arising in practice theoretically has at least a single infinity of natural frequencies, nevertheless in many cases a reasonably exact statement of the complete system may be made by considering only one of these frequencies at a time. The system then behaves nearly as a simple system comprising mass, stiffness and friction or damping. The response of such an idealized system to forcing at any frequency is commonly given in terms of these three parameters, but expressed in this form the equations are usually complex and their physical significance not easily comprehended except with practice.

The form recommended where convenience in calculation is valuable, expresses the steady response of the system at a given forcing frequency after transients have died away, in terms of the resonant response, namely the response when the forcing frequency is identical with the natural resonant frequency. It takes the form:—

$$u = \frac{F}{r} \cos\ (pt - \phi) \cos \phi$$

in which F is the magnitude of the sinusoidal exciting force and p its spin, the latter being a convenient term to express the angular velocity of the equivalent circular motion. The term r represents the friction or damping which is assumed to be of the viscous type in which the resisting force is proportional to the velocity of the system, and u is the velocity of the motion. The angle ϕ is given by—

$$tan\ \phi = Q \left(\frac{p}{\omega} - \frac{\omega}{p}\right)$$

in which ω is the spin corresponding to the resonant frequency. The factor Q has been called the dynamic magnification factor or sometimes the resonance factor since it is a measure of the increase of vibration due to resonance. It will be noted that the expression given above gives the *velocity* of the system from which, of course, the amplitude or acceleration may be derived by respectively dividing or multiplying by p. The factor F/r may also be written in either of the forms $QF/\omega s$ *or* $QF/\omega m$ as may be found most convenient, where m is the mass, s the stiffness and ω the resonant spin (given by $\sqrt{s/m}$).

This expression is particularly useful for rapid calculation for a number of reasons. In the first place a reasonable estimate of the magnitude of Q may generally be made in advance without any knowledge of the mass or stiffness of the system. For example, in a tuning fork, particularly a type carrying loading masses at the fork ends, the value of Q is usually very high between 1000 and 10,000. In an unbladed turbine rotor of the gashed type Q is usually of the order of 1000. When the rotor is bladed a value between 100 and 1000 is frequently obtained with a tendency for

low and variable figures before the rotor has been first run up to speed which assists in bedding down blade roots. Individual turbine blades or groups of blades may have a Q value in the region of 20 to 200 or higher. A riveted structure such as a bridge or machinery seating may show Q values of the order of 10—100 according to the quality of riveting. A torsional shaft system such as occurs in marine propulsion generally has a Q of the order of 8—20 and in such systems hydraulic damping at the propeller is the main reason for producing a low value of Q.

In the second place this expression shows at once one most important characteristic of simple forced vibration, namely, that the frequency of vibration is the frequency of *excitation* corresponding to the spin p and *not* the resonant frequency corresponding to the spin ω. This means that with a simple system, or a complex system in which one natural frequency is sufficiently far removed from the others to be relatively unaffected by their presence, the response to forcing must occur at precisely the forcing frequency. An exact determination of the actual frequency occurring can then indicate with precision what is the prime cause of the vibration even though precise knowledge of the resonant frequency is not available.

Thirdly, the maximum value of velocity occurring during any one cycle of the vibration (when only the exciting frequency varies) depends only upon the angle ϕ (which depends in a complicated manner upon the relationship between p and ω as well as upon the other constants of the system) and is given in a very simple form by the cosine factor $\cos \phi$. This has made possible the construction of a universal resonance nomogram [1] which is reproduced here (Fig. 1) by permission of the editor of *The Engineer*. This chart enables the response of a system of known characteristics subjected to forcing at any frequency, to be determined very quickly in the neighborhood of resonance. Far removed from resonance the response is simply calculated but in any event the vibrations which cause trouble are usually closely associated with the resonant condition.

A brief examination of this chart shows that for Q values higher than ten the response in the region close to resonance varies rapidly for small changes of forcing frequency. This is reflected in vibration measurements by the fact that the magnitude of vibration is rarely steady for more than a few seconds at a time, as is to be expected in rotating machinery where the speed of rotation may readily vary from the nominal value by one or two per cent. In the choice of equipment for detecting and measuring vibrations, it is therefore pointless to aim at a high standard of accuracy in the measurement of vibration magnitude, and normally an accuracy of $\pm 20\%$ is more than sufficient while much valuable work may be done with equipment whose calibration from the point of view of amplitude or intensity measurement is entirely unknown. On the other hand, the determination of frequency must be made to a very high order of accuracy, and this point will receive attention subsequently.

The information contained in the equation given above for the behavior of a simple resonant system may also be expressed graphically in another form. Fig. 2, which is reproduced here for the first time, shows a universal resonance chart which is particularly suitable for use when examining experimental measurements in order to ascertain whether they do in fact represent a simple resonant system and if so, to determine the appropriate value of Q.

In this case it is not possible to adopt a double scale for the mis-tuning since the two-dimensional character of the chart would result in confusion. Mistuning is therefore to be calculated as a percentage of the lower frequency, which will be the frequency of excitation when this lies below the resonant frequency. In the range above the resonant frequency the latter is to be used as the basis.

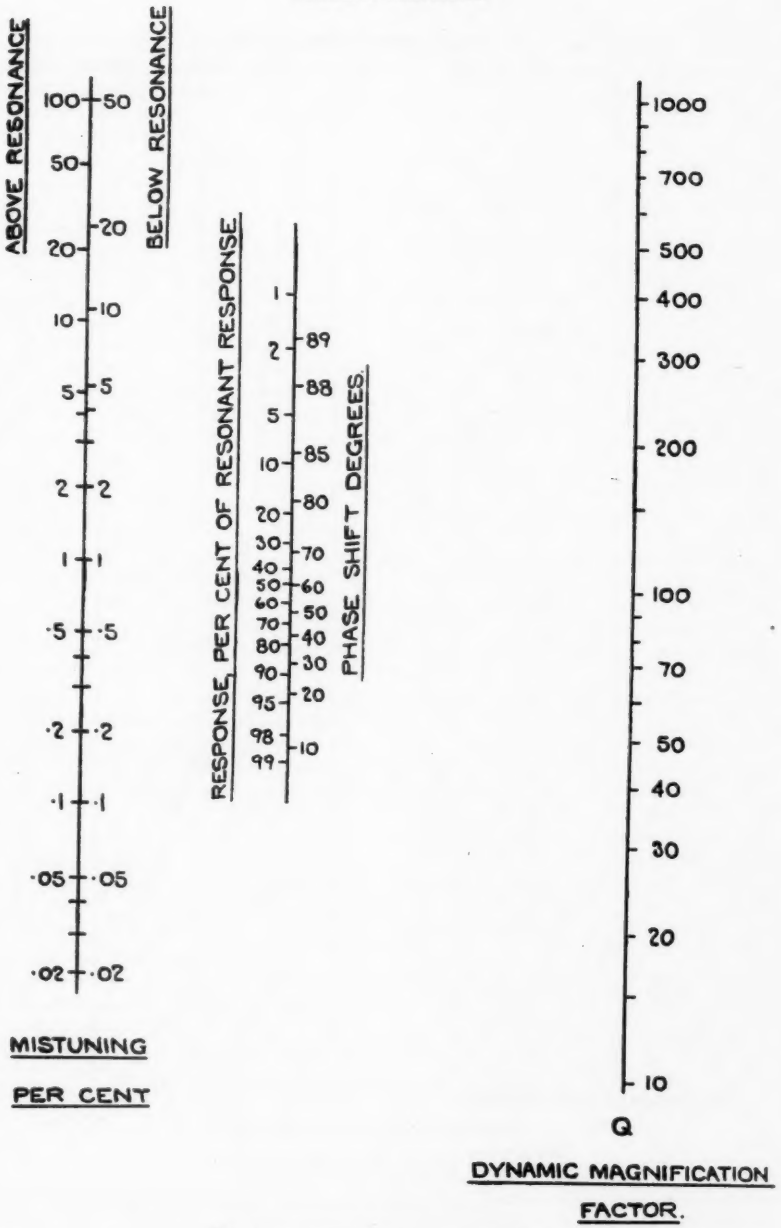

Fig. 1—Universal Resonance Nomogram

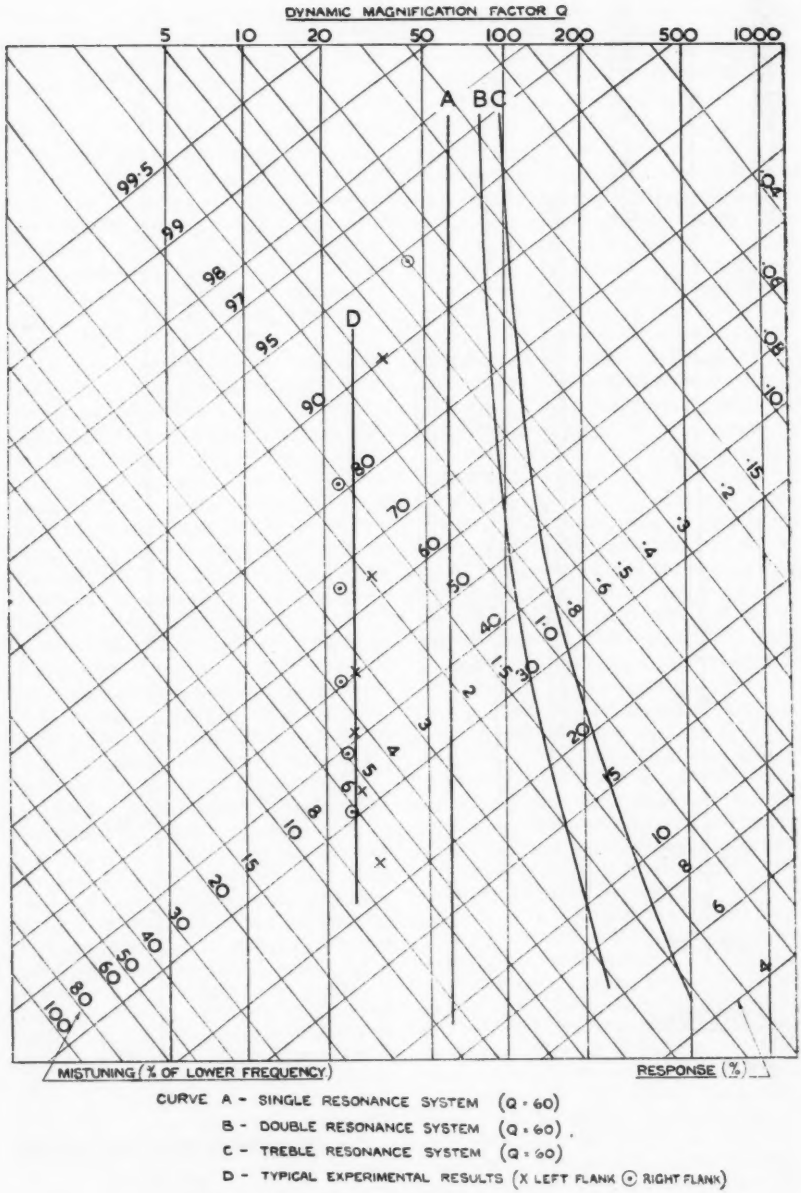

Fig. 2—Universal Resonance Chart

When the response expressed as a percentage of the resonant response is plotted against percentage mis-tuning on this chart, the points will lie on a vertical straight line if the system has simple resonant characteristics, and the position of this line depends on the Q value. As with the nomogram of Fig. 1, the scale is such that the region close to resonance may be examined in detail without requiring an exaggerated scale at points far removed from resonance. Lines marked A, B and C are included to indicate the extent of departure from a straight line when the system has double or treble resonant characteristics. That is to say, line A indicates the response of a simple system having a Q of 60, whereas B shows the behavior when two such systems are used in cascade with the excitation for the second system derived from the first. The resonant frequency is identical for both, i.e. there is no separation such as would produce a band-pass effect. It will be noted that in the immediate vicinity of resonance the line is nearly vertical and corresponds to a Q greater than 60 in the ratio $\sqrt{2}$: 1, but at points further removed from resonance the line curves off towards the right-hand side of the chart. In the treble resonant system involving three simple systems all having a Q of 60 and the same resonant frequency, the behavior near resonance approximates to that of a simple system with a Q of $\sqrt{3}$ times 60, and further away from resonance the line again curves off to the right and more sharply than for the double system.

To illustrate the value of this chart for the purpose for which it is intended, line D is drawn with reference to the marked points which, in fact, represent experimental measurements made during the analysis of the natural vibration characteristics of a second-reduction gear wheel. Prior to the preparation of this chart, it was necessary to plot such measurements on logarithmic paper, estimate by inspection the probable value of Q, and draw in, point by point, a resonance curve to see if it coincided reasonably closely with the measured points. With this new chart no such trial and error process is necessary. For practical purposes the number of reference lines on the chart is, of course, increased, and many of these have been omitted in Fig. 2 for the sake of clarity.

The foregoing remarks relate to the steady state response of a simple vibrating system after initial transients have died away. Such transients are seldom of importance in vibration associated with running machinery but they have an important effect on the behavior of instruments designed to measure such vibration. When an alternating exciting force is applied to a vibrating system its effect increases at first rapidly and then at a reduced rate, so that the final response is attained in exponential fashion and the time to reach any desired fraction of the ultimate steady state response depends on the frequency of excitation and on the value of Q. This effect may be most conveniently expressed in terms of the number of cycles required to reach approximately full response. Regardless of frequency a response of 96% of maximum will be achieved after Q cycles of the excitation have been allowed to act on the system. For example, a system having a Q of 100 and stimulated at a frequency of 1000 cps will attain practically full response in a tenth of a second, whereas it would require ten seconds to do so at the much lower frequency of 10 cps. Any small variation in exciting frequency occurring within the time required for build-up is therefore much more important at low than at high frequencies.

THE VELOCITY CRITERION

For some reason it is customary to express the magnitude of mechanical vibration in terms of its linear or angular amplitude. This has presumably arisen because it is the only parameter immediately perceptible by the human

senses, and it dates from the days when vibration measuring equipment was not available and the engineer had to rely on visual measurements of the actual amount of motion taking place. It will be obvious that the amplitude of vibration permissible in a large low-speed machine would be quite prohibitive in a small fast-running machine of otherwise similar design. For example, specifications for turbines used in land power stations sometimes call for a guarantee that the amplitude of pedestal vibration shall not exceed 0.002 in. While this is relatively easily achieved in machines running at 3000 r.p.m., it would be difficult to ensure in a 1500 r.p.m. machine, and would be no guarantee of smoothness in a high-speed turbine running at 6000 r.p.m.

It is obviously desirable to measure vibration magnitude in a unit which will result in similar values for all machinery of the same type and constructed with the same degree of precision. It might appear that the maximum acceleration occurring in the vibration would be a suitable criterion, since this is a measure of the forces acting upon every pound of that part of the system which is concerned in the vibration, but a little consideration will show that this criterion has erred in the other direction, because the larger and therefore heavier low-speed machine will comprise parts whose masses vary with the cube of the linear dimensions while the cross-sectional areas available for carrying loads only increase as the square of these dimensions.

There are a number of reasons for concluding that the velocity of vibration is the most suitable criterion, and these are given in detail in Appendix 1. What has just been stated appears reasonable on qualitative grounds and is in fact borne out in practice.

Measurements of bearing pedestal vibration made in a large number of ships on the basis of vibration velocity (in which the rms value of velocity was recorded) indicate that Table 1 represents a reasonably accurate relation between vibration velocity and smoothness of running. The column headed "Equivalent db" will be referred to subsequently. The measurements were made with a direct-reading portable vibrometer developed for the purpose.

The use of velocity as a basis is already accepted in noise measurement, where the noise intensity is defined as energy flux per unit area of cross section and is proportional to the square of the air particle velocity (neglecting variations in atmospheric density). It

TABLE 1

Condition	*Ins/sec. rms.*	*Equivalent db.*
"Spins like a top"	.05	90
Very good	.1	
Normal	.2	100
Slightly rough	.5	110
Unsatisfactory, but probably not immediately dangerous	1	
Immediate investigation desirable	2	120

is, of course, true that at any one frequency noise intensity is proportional to the square of any of several parameters, e.g. sound pressure, particle displacement, velocity or acceleration. However, apart from pressure, the velocity is the only true measure of sound intensity when the frequency also varies.

The third column of Table 1 indicates the sound level in decibels of plane sound waves having rms. particle velocities equal to those given in the second column. Conditions in an engine-room do not correspond precisely to plane waves, and one would expect that the sound intensity measured at some distance from the source of disturbance would be less than that having the same particle velocity at the source, particularly if the source is small, as for example a bearing pedestal in contra-distinction to a large surface such as the wall of a gear case. However, it is clearly more than a coincidence that the value of bearing pedestal vibration velocity referred to as "normal" in Table 1 corresponds to a noise level at the source of 102 db. The standard of smooth running chosen in this Table was derived as already explained by many measurements on running machinery, and, therefore, may be taken to represent average good practice. On the same basis the noise intensity in an engine-room having machinery of good design and built to modern standards of accuracy is usually of the order of 100 db. Similarly, the velocity of 2 in. per second, suggested as an upper limit demanding immediate investigation, corresponds on the basis of equivalent particle velocity to a noise level of 120 db which will be generally accepted as a very high noise level and would, of itself, require action to be taken.

The stress induced in the structure of a machine by vibration velocities of the order of one or two inches per second is not of itself sufficient to cause failure of the part having these velocities, and their significance lies in the fact that fortuitous resonances of some other part may cause much larger vibration velocities elsewhere. For example, a pedestal vibrating at 1 in/sec. may only be stressed thereby to about an eighth of a ton per square inch, but a turbine blade excited to resonance by the same vibration may readily be stressed up to five or ten tons per square inch, which may have serious consequences, particularly if the blade already carries centrifugal stresses of some magnitude.

Table 1 relates to turbine and gearing bearings in main propulsion machinery, and represents in the Author's opinion a reasonable comparative basis for such plant. It is not suggested that the same scale necessarily applies to machinery of very different type, as for example, a railway engine or a sewing machine, since the limiting factors may be questions of habitability and comfort rather than mechanical stresses and also because the relationship between vibration velocity and stress necessarily varies somewhat in machines of different type. However, this variation is less than is commonly supposed, as will be seen from Appendix I.

The choice of vibration velocity as a criterion has a further advantage from the point of view of the design of instruments to measure and record vibrations. In one of the simplest types of vibration pick-up, a magnet is attached to the vibrating body and has suspended within its field a coil of wire attached by a light suspension. The natural frequency of the coil on this suspension is made sufficiently low to be below the lowest frequency which it is desired to record, and in such circumstances the coil remains approximately fixed in space while the magnet moves past it with a motion equal to that of the vibration under investigation. The voltage output from the coil is directly proportional to the velocity of vibration and therefore an instrument of this type automatically reads velocity. In the crystal type of

pick-up as normally employed, the potential difference set up across a crystal is proportional to the acceleration of the motion and such pick-ups tend to give a low sensitivity for vibrations of low frequency. If they are followed by an integrating stage, the resulting output is once more proportional to velocity for a wide range of frequency whose lower limit depends on the time constant of the integrating circuit. At first sight it might appear that the crystal pick-up has a great advantage in that it does not require a seismic mount but is attached directly to the vibrating body, whereas the moving-coil type has a natural resonance which must lie below the lowest frequency to be dealt with. This is sometimes inconvenient since limiting this resonance to a very low frequency means necessarily a relatively large mechanical displacement under the weight of the coil, so that it is inconvenient for use in both vertical and horizontal directions at the same setting. However, the advantage of the crystal is more apparent than real since the use of a long time-constant in the integrating circuit, while it may allow the available frequency range to be extended downwards, nevertheless at the same time brings about a reduction in sensitivity precisely proportional to the lower limit of frequency at which constancy of proportionality is still achieved.

CONVENIENCE OF NOISE MEASUREMENT

The opinion is sometimes expressed, and still more frequently is held without conscious expression, that the correct way to approach a vibration problem is to measure the mechanical vibrations which are present. It is certainly true that this must be done if a complete statement of the conditions is to be made, but if the prime object of the investigation is to trace the origin of the disturbance so that it may be eliminated, then there is much to be said for doing at least the preliminary work by measurement of noise. This is, of course, useless at very low frequencies since neither the ear nor a microphone of normal type has appreciable sensitivity for frequencies lower than about 60 cycles per second. Nevertheless there remains in marine engineering a wide range of vibration phenomena in which the frequencies are higher than this lower limit, and in such cases the microphone is an extremely convenient way of picking up sufficient of the disturbance to identify its source. The microphone does not require physical attachment to the machinery, and in the confined space of a ship's engine-room noise emananting from any part of the machinery may be transmitted to other parts of the engine-room with relatively little attenuation. Sometimes, of course, it may appear to gain in intensity by such transmission due to the setting up of standing waves in the air, but these do not usually cause serious difficulty. In any case, mechanical vibration in a structure may also be transmitted to other parts of the structure with enhancing of its intensity at certain points due to local resonance. As mentioned earlier, the intensity of most vibrations which actually occur and give trouble in marine engineering installations generally varies from moment to moment as well as from place to place, so that an exact measurement of magnitude is both difficult and pointless.

The equipment to be described for analyzing and interpreting vibration measurements may be actuated by any type of microphone or vibration pick-up, whether adapted to measure linear or angular motions and whether attached to pedestals and seatings or to rotating parts. The pick-up is merely a means of translating bodily motion into an electric voltage, containing in its variations all the characteristic qualities of the physical motion and in particular all its component periodicities.

WAVE ANALYSIS

It rarely happens that objectionable noise or vibration consists so largely of one frequency component that a precise diagnosis may be made by one simple frequency measurement. In general the disturbance is very complex, containing so many components covering a wide frequency range that even the strongest of them cannot be identified with certainty without some means of separating it from the others. This process of separation is given the general name of "wave analysis," and the operation may be performed in a variety of ways.

Perhaps the oldest method is to secure a visible record, on paper, celluloid or photographic film, of the form of the vibration or noise, which shows how the quantity measured varies with time. The numerous methods of obtaining such a record have been very fully described elsewhere, and will not be dealt with in detail here, except to say that for low frequencies (say up to several hundred cycles per second) the mechanical vibrograph recording on celluloid is very convenient, while photographic records may be made up to 10,000 cps. or higher. The record must then be examined by the methods of Fourier analysis, which enables the strength of the various harmonics of the fundamental frequency to be determined with greater or less accuracy according to the complexity of the record. In practice, noise and vibration occurring in running machinery are usually so complex that the basic frequency, of which all higher components are to be expressed as multiples in the Fourier method, is extremely difficult to identify since it has really no physical significance. For this type of record the envelope method described by Manley [2] is particularly suitable.

Fourier analysis of a complete wave form is a very lengthy business, and cannot possibly be done on the spot. Records must be taken away for examination and analysis, and if further information is found to be necessary, this requires a further trial, which is not always convenient for the shipbuilder or shipowner. Much attention has, therefore, been devoted by many workers during the last fifteen years to the development of instruments which will give a direct analysis on the spot, enabling further measurements to be quickly made in order to check a tentative theory.

These wave analyzers may be broadly divided into two types, which differ in one very fundamental way. In the first type the sole indication of the presence of a component of any given frequency is given by a meter reading, while the second type acts as a tunable filter, passing on the selected component as a sinusoidal variation of, usually an electrical voltage, and suppressing other unwanted components. Analyzers of the first type lose the oscillatory character of the particular component whose existence they demonstrate and whose strength they can measure. Only the tuned filter type of analyzer can retain the oscillatory character of the wanted component, and experience has shown that it is most important to be able to do so.

Appendix III explains in more detail the operating principles of various types of wave analyzer and their relative advantages and disadvantages. Research work at Pametrada Research Station has resulted in the development of improved forms of wave analyzer of the tuned filter type and Fig. 3 shows the degree of selectivity actually obtained with these instruments together with the performance of an analyzer employing the simple Scott bridge. In addition to improved selectivity the Pametrada wave analyzers incorporate a selectivity control by means of which the frequency responsive character of the instrument can be varied, even down to the limit of zero selectivity at which the whole input signal is transmitted without fre-

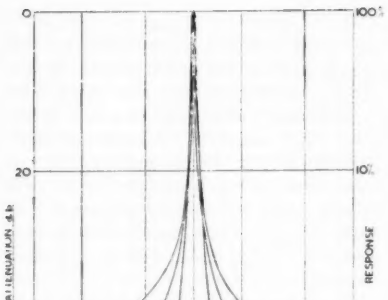

Fig. 3—Wave Analyser Response Curves

quency discrimination. In practice this is found to be a most valuable feature and the instrument combines the functions of wave analyzer and sound level meter.

Identification of Source—the Wave Correlator

Having isolated any one component from the complex wave, it is next necessary to identify its origin if the analysis is to serve any practical purpose. For this a measurement of frequency, which is immediately available from the calibrated scale of the analyzer tuning control, is very useful and often sufficient. But there is generally some doubt about any deductions made merely from measured frequencies, since both the analyzer scale and the scales of shaft tachometers are liable to small errors of calibration or reading, and furthermore, they cannot be read at precisely the same instant, so that the engine speed may have drifted slightly in the meantime. This is well illustrated by Figs. 4 and 5, in the first of which only analyzer frequency readings and shaft tachometer indications were available. The plotted points lie sufficiently near to lines indicating certain multiples of shaft speed as to suggest that the observed frequencies were probably those multiples. However, in the second graph special steps

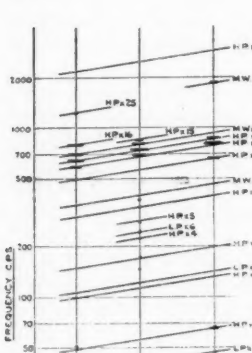

Fig 4

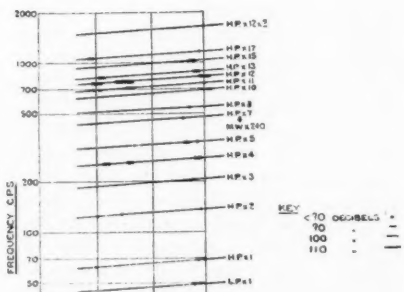

Fig. 5

Charts of Noise Frequencies Induced by Flexible Coupling (Example 1)

were taken to ascertain whether this was in fact the case. It was found to be so, and the main shaft speeds plotted as abscissae were then calculated from the frequency ratio. The trials to which this graph refers were taken at three selected nominal speeds, and the plotted readings show that the speed actually deviated from the nominal values by a small percentage.

In addition, then, to mere measurements of frequency there are two methods of identification capable of greater exactness. The first, which is by far the less powerful of the two and is only occasionally applicable, is to look for some particular *quality* in the component under investigation and then seek for some part of the machinery which also possesses that quality. This may take the form of a modulation at some lower frequency, so that the amplitude of the vibration or noise component rises and falls at a definite and measurable periodicity. This is illustrated in Examples 2 and 3.* Of course a high-quality analyzer will do a great deal to remove such modulation, so that this method of identification finds its greatest application in rough-and-ready diagnosis with very elementary apparatus, as was the case in both examples quoted.

The second method of identification is much more refined and powerful, and often gives results which permit attention to be withdrawn from every part of the machinery except one, and concentrated on that one with a certainty which cannot be gainsaid. A certain knowledge that the trouble lies in a restricted and identifiable part of the machinery is often half way to the solution.

This method consists in comparing the component frequency with the rotational speed of each shaft in the machinery or in any of its auxiliaries, by a technique which is capable of detecting differences of the order of one part in many thousands. In its simplest form (described in Example 1) a cathode-ray oscillograph, which may conveniently be of the double-beam variety, is used. The time-base or horizontal sweep is adjusted so as to recur slightly less frequently than the rotation of the shaft, and a timing pulse derived from the shaft itself is applied to the synchronizing terminal so as to return the beam to the beginning of its sweep at precisely each revolution of the shaft. The timing pulse may be generated by a small magnet carrying a pick-up coil and fixed near the shaft, which is provided with a small notch or projection at one point on its periphery. If a double-beam tube is used, the timing pulse may be applied to the input side of the appropriate amplifier and the synchronizing connection taken from the deflector plate of the second beam. This has a double advantage in that an amplified voltage is available for synchronizing, and also the pulse is made visible on the second beam, which will demonstrate that the sweep frequency is equal to the shaft speed and not a fraction nor a multiple of it.

The vibration or noise component under investigation, after selection by the analyzer, is now applied to the deflector plates of the main or first beam. If its frequency bears no integral relationship to the speed of the shaft, then the pattern on the oscillograph screen will appear to be moving laterally, possibly so rapidly as to cause only a general blur. If, however, the frequency ratio is a whole number, then the pattern will appear to be stationary, having one complete wave for each unit of the integral ratio. Even a very small departure from an integral value will eventually cause a noticeable shifting of the pattern. For example, suppose that a frequency of approximately 720 cps. is being compared with a shaft speed of 60 revolutions per second or 3600 r.p.m., twelve complete sine-waves will appear, and if the pattern does not move laterally by more than half a wave in, say,

* For examples see p. 529, et seq.

two minutes, then it is clear that the frequency ratio cannot be greater or smaller than twelve by more than one part in about 170,000. Experience (Example 1) shows that in such a case the order of accuracy quoted is easily obtainable.

The method described above is only applicable provided that two conditions are fulfilled. The first of these is that the ratio is not too high. For any integral multiple up to about twenty or in exceptionally favorable circumstances perhaps forty, the time base of the oscillograph may be triggered by a pulse derived from shaft rotation and the appearance of a stationary pattern on the screen may be definitely identified if the number of complete waves visible does not exceed the values mentioned. Clearly the higher values can only be recognized if the component is very steady in amplitude. If, however, the ratio is much larger, such as would result from the attempt to compare the tooth-engagement frequency with the speed of rotation of the low-speed wheel of a gear train, then the pattern becomes so fine in texture that the eye and screen are incapable of resolving it.

Some improvement can be obtained by broadening out the horizontal trace so that only a fraction of it is visible on the screen but in many cases the second condition then becomes vitally important, namely that the frequency of repetition of the complete pattern must be sufficiently high to enable the eye to appreciate the pattern as a whole instead of following the movement of a point of light. A long delay screen is of some assistance but can only affect limited improvement.

A new instrument which has been called the "wave correlator" provides a more precise means of recognizing integral frequency-multiples with sufficient discrimination to be able to deal with multiples up to the 600th order or thereabouts. It incorporates four separate channels so that frequency relationships with four different shaft speeds may be examined simultaneously. Each channel has two neon tubes controlled both by the vibration component under investigation and by the shaft periodicity. One tube flashes if the vibration signal is positive at the moment when the timing pulse arrives from the shaft, and the second flashes if the signal is negative at the pulse instant.

When the signal frequency is an integral multiple of shaft periodicity, the pulse always arrives with the same phase relationship to the signal and so only one tube of the pair will fire. For non-integral frequency ratios both tubes will fire at different times, and if the shaft speed is sufficiently high their illumination will appear continuous.

The correlator is fully described in Appendix III.

"VEST-POCKET" INSTRUMENTS

The precise and powerful instruments described in the foregoing sections are all portable. They have been designed to facilitate use on shipboard, and have been so used on numerous occasions. Nevertheless, although each piece is readily portable, some consideration must be given to transport since there are four essential items (analyzer, analyzer power unit, correlator and oscillograph), in addition to microphone, pick-up and timing devices, headphones, spare valves, and so forth, and frequently a small motor-alternator set to obtain the necessary supply of 250-volt alternating current. Consequently, the full equipment is usually reserved for cases where serious vibration or noise is known to be present.

Experience has shown that certain instruments of really portable type can be of very great value, even if much more elementary than the full analyzing and correlating equipment. A good deal of attention has been paid to the selection of pocket instruments, so small and

light that the observer is not inconvenienced by carrying them with him on every trial, even if there is no prior reason to suspect that they may be required.

Reed Vibrometer

One of the most valuable instruments of this type is the tuned-reed vibrometer, of which the type known as the Anson is particularly convenient. It consists of a steel tape wound round a drum concealed inside the body of the instrument, and protruding through guides. A knurled knob permits the drum to be rotated so as to vary the projecting length of tape, and a calibrated scale enables the natural frequency of the reed so formed to be read off. In use the body of the vibrometer is held against the machinery under test in such a way that the vibration is approximately at right angles to the plane of the reed. The knurled knob is turned slowly and steadily through the tuning range, and when any one component of the vibration is attained the reed will vibrate comparatively strongly. The range of frequencies available on the calibrated scale extends from 200 to 20,000 per minute and the effective Q of the analyzer is in the neighborhood of 50 to 100. Some care is required at the very low frequencies, since the time to build up maximum response is naturally rather long in this region and unsteady holding may set up transient vibrations which take so long to die down that they may be mistaken for forced resonance. The instrument cannot, of course, be applied to very small parts, since its weight in contact with them might destroy the vibration or modify its frequency seriously, but for turbine pedestals, platforms, gear cases, thrust blocks and so forth, it is very convenient. The instrument is also provided with a detachable spindle carrying a slightly eccentric weight and terminating in a rubber "center" which may be applied to the end of a rotating shaft. The unbalance force acting on the weight as it revolves will actuate the reed when it is tuned to resonance with the speed of rotation of the shaft.

In using this instrument the reed will sometimes vibrate in loop form with one or more nodes along its length. This means that it is moving at one of its higher orders of vibration. Such indications must be disregarded and a search made higher up the frequency scale with a shorter reed, until a true cantilever motion is found. The ratio of true frequency to spurious frequency may be estimated from the data in Fig. 6 for the clamped-free mode of transverse vibration of a rod.

NATURAL FREQUENCY = $0.391\,K\frac{m}{L^2}$ C.P.S.

WHERE K = RADIUS OF GYRATION OF SECTION, INCHES. L = LENGTH OF BAR, INCHES.
C = VELOCITY OF SOUND PROPAGATION, IN/SEC. m = NUMBER DERIVED FROM TABLE BELOW

MODE OF SUPPORT	VALUES OF m					VALUES OF m X AT NODES
ORDER	1ST	2ND	3RD	4TH	ETC	(NOT TO EXCEED m)*
CLAMPED - FREE	1·19	3	5	7		0 $\frac{22}{100}$ $\frac{78}{100}$ ETC
CLAMPED - PARALLEL	$1\frac{1}{2}$	$3\frac{1}{2}$	$5\frac{1}{2}$	$7\frac{1}{2}$		0 $\frac{25}{100}$ $\frac{75}{100}$
PINNED - PINNED	2	4	6	8		0 2 4
CLAMPED - PINNED	$2\frac{1}{2}$	$4\frac{1}{2}$	$6\frac{1}{2}$	$8\frac{1}{2}$		0 $\frac{39}{100}$ $\frac{61}{100}$
CLAMPED - CLAMPED	3	5	7	9		0 $\frac{50}{100}$ $\frac{50}{100}$
FREE - FREE	3	5	7	9		$\frac{1}{2}$ $\frac{22}{100}$ $\frac{78}{100}$

NODES OCCUR AT DISTANCES X L FROM ONE END WHERE THE POSSIBLE VALUES OF X ARE OBTAINED BY DIVIDING THE APPROPRIATE VALUES OF m X GIVEN IN THE TABLE, BY THE VALUE OF m FOR THE PARTICULAR ORDER UNDER CONSIDERATION.

* THE MAXIMUM VALUE OF m X FOR THE CLAMPED - CLAMPED MODE IS $\frac{1}{2}$, AND FOR THE CLAMPED - FREE AND FREE - FREE MODES IT IS $\frac{1}{2} - \frac{1}{4}$ (AND NOT $\frac{1}{2} - \frac{1}{2}$ AS GIVEN IN THE TABLE)

EXAMPLE :- FOR THE FOURTH ORDER OF THE CLAMPED - FREE MODE THE NODES OCCUR AT THE FOLLOWING VALUES OF X

0, $\frac{39}{100}$, $\frac{61}{100}$ AND $\frac{78}{100}$

i.e. 0, $\frac{2}{5}$, $\frac{3}{5}$ AND $\frac{4}{5}$

APPROXIMATE VALUES OF C FOR COMMON MATERIALS (m/SEC × 1000) :-

STEEL. 200 ALUMINIUM. 200
BRASS. 135 - 155. CAST IRON. 135 - 140

Fig. 6—Transverse Vibration of Uniform Bars

Pitch Pipe

For the determination of higher (audible) frequencies a pitch pipe or tunable whistle is very helpful if properly used. This is simply a whistle having a tuned reed or a movable piston to vary the note. One in the Author's possession is of the former type, and is consequently very compact, being only 2½ in. long overall. It is blown (or rather sucked) by the mouth, and the note varied by a calibrated cam. Although it only covers a range from 360 to 780 cps., it has nevertheless been found of great value when once the proper method of employing it had been discovered. First attempts were completely unavailing, as the note produced by the pipe was far too weak to be audible in any ordinary engine-room, much less in a noisy one. However, the use of a stethoscope applied to the body of the pipe reduces the engine noise heard by the ear and at the same time picks up the pipe note more strongly. The pipe is then clearly audible, even when blown gently which gives the purest note. If the engine noise has only two or three predominant components, their frequencies may be accurately located by the presence of a slow beat when equality of pitch is nearly reached.

Stethoscope

The ordinary medical stethoscope is not of much use in studying engine vibrations, but it may be much improved by removing the vulcanic cap and replacing it by a rubber bung having a small hole bored through the center. If a metal rod is inserted in the hole, of a size to fit it tightly, vibrations in machinery are rendered clearly audible by touching the end of the rod against an appropriate part of the machinery, and at the same time general air-borne noise is transmitted only weakly. By removing the rod the instrument may be used to search for localized sources of air-borne noise.

Acoustic Resonators

Preliminary work has shown that simple Helmholtz resonators, of the tunable type with a variable cavity, are likely to prove of considerable value. While not in themselves sufficiently selective to give accurate frequency determinations, they will increase the relative intensity of a weak component sufficiently to obtain the frequency accurately by beating with the note of a pitch pipe. Further work on convenient forms of resonator is proceeding.

SIGNIFICANCE OF CERTAIN HARMONIC PATTERNS

The presence of a strong component having a frequency related to the speed of a particular shaft by an integral multiple enables attention to be directed to that shaft, or another running at precisely the same speed, to ascertain what mechanical feature is capable of causing regularly repeated disturbances exactly so many times per revolution. For example, the number expressing the multiple may be the number of teeth in a gear wheel or pinion carried on the shaft, or the number of teeth in the master wheel of the hobbing machine which was used to cut the gear. It may be the number of bolts in a flanged coupling running sufficiently fast to cause an audible noise or a simple multiple of that number. The possibilities are numerous, and each case must be examined on its merits. There are, however, certain multiples, or combinations of multiples, which arise sufficiently frequently to warrant specfal mention.

First Multiple

The most usual cause of a disturbance arising once per revolution is eccentricity in one form or another. Unbalance of a turbine motor or pinion will

give rise to this effect, and its presence can usually be detected in high-speed machinery even if the balance is good.

Second Multiple

The presence of an appreciable amount of second harmonic points to some measure of ovality in the shaft, either literally or effectively. The most common cause of second harmonic is a bent shaft. Even if the shaft is re-balanced so that the first harmonic (*i.e.*, once per revolution), disturbances are more or less completely removed, the journal will be forced to rotate about an axis which is approximately the straight line joining the centers of both journals, and does not coincide with the geometrical axis of the journal. Consequently, the journal will execute an "elbowing" motion in its bearing. The load will only be distributed more or less uniformly along the length of the bearing at two particular orientations of the shaft, and in between these two positions there will be concentrations of load first at one end of the bearing and then at the other end. Thus forces arise having a periodicity of twice the speed of shaft rotation. This condition arises much more frequently than is generally supposed, and only requires a very small bend to give an appreciable effect. The thickness of the oil film in a bearing is so small that even a difference of one thousandth of an inch between the film thicknesses at the two ends of the bearing is sufficient to cause a considerable disparity in the loads carried at the two ends. In practice a bend of 0.002 in. in the length of the bearing may cause distinctly rough running. The effect may be aggravated if the pedestal has any tendency to rock as the load on it travels axially back and forth along the length of the bearing. Incidentally, the frequency of such rocking motion will be equal to, and not twice, the shaft speed as the successive load peaks act alternately at opposite ends of the pedestal.

The fitting of a spherically seated bearing does nothing to overcome the elbowing action in a high-speed bearing, since the spherical seat is only adapted to correct the static alignment of journal and bearing and will be prevented by friction from oscillating at shaft-rotation frequency.

Multiples n-1, n, n + 1

If a sinusoidal disturbance occurs n times per revolution of a shaft rotating with angular velocity p radians per second, it may be expressed as a function of time in the form of $a \cos (npt + \delta)$, where a is the half amplitude and δ is a phase angle which may be measured from any suitable datum position of the shaft. (Incidentally, if the disturbance is not sinusoidal its expression will contain harmonic terms of spin $2np$, $3np$ . . . etc., as well).

Now it may happen that the amplitude a is not constant, but varies slightly in a cyclical manner as the shaft revolves, having one maximum and one minimum value during each revolution. For example, a gear wheel of n teeth may produce torque variations or audible pulsations n times per revolution, and in addition the tooth annulus may be set slightly eccentric to the shaft axis (see Example 2). In this case the amplitude of the disturbance will not be constant, but will take the form $a + b \cos pt$ (if the phase datum is chosen suitably). Hence the disturbance, expressed as a function of time, will contain the term $b \cos pt \cos (npt + \delta)$ in addition to the nth multiple $a \cos (npt + \delta)$. This additional term involving the product of two cosines represents a modulation of the nth multiple by the first multiple of shaft periodicity, and may be expressed by simple trigonometrical rules in the form

$$\tfrac{1}{2} b \left\{ \cos \left[(n-1)pt + \delta \right] + \cos \left[(n+1)pt + \delta \right] \right\}$$

It will be observed that this represents two new multiples, the $(n-1)$th and

$(n + 1)$th, each having a half amplitude of ½b.

These components have a real existence and may be detected by a wave analyzer if it has sufficient selectivity to distinguish between the multiples $n - 1$, n, and $n + 1$. The identification of three successive integral multiples may, of course, be due to the actual presence of three separate disturbances, but in practice it is much more frequently caused by a modulation of the middle one of the three by the frequency of shaft rotation, as explained in the preceding paragraphs. In such a case the intensities of excitation of the multiples $n - 1$ and $n + 1$ are equal. However, the observed noise or vibration may be due to the response of one or more resonant systems to this excitation, and if so the observed strengths of the two components may be unequal, since a resonant frequency may be closer to one of the two component frequencies than to the other. There is also another way in which the actual intensity of excitation may be different for the two multiples, namely, if there are present two different modulations, each at rotation frequency but in different phases. For example, a double-ended flexible coupling of the toothed type, having n teeth or claws, will produce a disturbance at n times rotation frequency if the shafts are even very slightly out of alignment, since the load will no longer be evenly distributed among the teeth. If in addition one toothed element is slightly skewed on its shaft, the unevenness of load distribution will vary slightly once per revolution and the resulting modulation of the nth multiple will give rise to additional multiples $n - 1$ and $n + 1$ of equal strength. If, however, a second toothed element at the other end of the coupling is also skewed, and in a direction suitably related to the skewing of the first, then the strengths of the sum and difference multiples will be unequal. (See Example 1).

With toothed couplings and shaft misalignment, particularly where the driving flanks of teeth or claws are straight, *i.e.*, without crowning, the above analysis is over simplified and many other multiples can arise, *e.g.*, $2n \pm 1$, etc.

DIAGNOSIS AND TREATMENT

No hard-and-fast rules can be laid down, since the possibilities are so numerous. However, the methods described in the foregoing sections have been found very useful for identifying the *cause* of unwanted noise or vibration. If the vibration is also accompanied by mechanical failure of certain parts, *e.g.*, turbine blades, it is then necessary to ascertain whether these parts have natural resonances at any frequencies found to be present in strength at any particular speed in the operating range. If this is the case, the probability that the failure is caused by vibration is strong. The subsequent course of action must depend on the circumstances of the case.

Several possibilities present themselves. The most satisfactory cure is to remedy the fault which is causing the vibration. However, this may be difficult and expensive, as for example re-hobbing one or more gear wheels or pinions. It is therefore frequently necessary to consider other alternatives, even of a temporary nature.

A particular row of turbine blades may be replaced by a stiffer section, or provided with additional binding wires, in order to avoid a particularly strong excitation frequency by raising the blade natural frequency until it lies above the highest excitation frequency experienced in the operating speed range. Alternatively, the operating schedule may be changed, so as to avoid any speed to which the blade row is particularly susceptible. This, however, runs the risk of merely transferring the trouble to

another row of blades of different natural frequency.

If blade failures occur *without* the accompaniment of heavy noise or vibration at frequencies near to one of the blade natural frequencies, then the possibility of self-excitation, due to unequal distribution of steam flow from the preceding row of nozzles or fixed blades, must be examined. This is outside the scope of the present paper, and it will suffice to say that such self-maintained vibrations can usually be prevented by attention to such factors as nozzle design, presence of shrouding or binding wires, number of blades under one length of shrouding, and accurate machining of blade roots.

It should be noted that in vibration problems such as most commonly arise in geared marine turbines, there is a sharp distinction between low frequencies of the order of one to ten per second and the higher range above about 30 per second. In the low-frequency range the technical difficulties are much reduced since *ipso facto* the spectrum cannot contain many harmonics of fundamental frequencies and furthermore the human senses can do a great deal of analysis without assistance by elaborate instruments. This is fortunate since analyzers of high resolving power are difficult to construct for very low frequencies, and furthermore the transient effects due to variation of frequency make it difficult to use a high resolution. The low-frequency range in marine engineering is generally concerned with oscillations of shaft systems (whether torsional, longitudinal or transverse) and with vibrations of the ship as a whole. The study of such problems has already reached a stage at which equipment and technique are adequate to deal with them, partly for the very reason given above, namely, that the problem is inherently simpler. The examples of vibration problems which follow are therefore restricted almost entirely to the high-frequency range.

EXAMPLES OF VIBRATION PROBLEMS

Vibration phenomena in marine turbines can take so many forms that the best way to illustrate the principles described in the foregoing paragraphs is to give a brief summary of a number of cases which have actually occurred during the last one or two years, and in which some or all of the instruments already described have proved of service in ascertaining, and in some cases removing, the cause of the trouble. It can scarcely be over-emphasized that successful diagnosis of vibration problems requires first of all much hard work, patience and practice. An experienced observer with very elementary equipment can often make a more satisfactory diagnosis than an unpracticed individual, even assisted by the most elaborate instruments.

The descriptions of the various examples given below are necessarily brief and their length may bear no relation to the actual amount of work which had to be carried out in each case. For instance, Example 1 involved seven separate trials on four different ships covering a period of nine months and a great deal of analytical work in addition. On the other hand, the analysis described in Example 3 was completed in less than an hour. The descriptions set out the relevant facts in order of their relationship and not necessarily in the order in which the observations were made, since the latter is usually conditioned by availability of machinery for trials, possibly of opening up for internal examination and testing, and other such factors.

Example 1

Several vessels of a class of which considerable numbers have been built were found to suffer fractures of h.-p. turbine blading in the fourth expansion,

the clear blade height being about 2 in. In some cases the failures occurred after the blading had been replaced and usually during the first subsequent voyage. The gearing was described as very noisy in all the installations affected, the noise being of high pitch and very penetrating. The failures appeared to be confined to a group of vessels, though at the outset no difference between these and other vessels of the class was apparent.

Measurement of the natural frequencies of the affected blades indicated a fundamental frequency of approximately 820 cps. the next higher frequency being approximately 1600 cps. The blades of the next expansion had sometimes failed (though this might have been due to damage resulting from failure of the previous rows) and had a functional frequency of approximately 540 cps.

Noise and vibration measurements were made using the wave analyzer in vessel *A*, and the result of the noise analysis is plotted in Fig. 4. This shows the observed component frequencies plotted to a base of main-shaft r.p.m., the noise levels being indicated by the widths of the short horizontal lines used to indicate each observation. Sloping lines are also drawn to indicate integral relationships between component frequency and the speed of certain shafts in the system. The installation comprised h.-p. and l.-p. turbines driving through double-reduction gearing of the interleaved type, the primary wheels and secondary pinions being of identical dimensions for each of the two trains.

It will be observed that the strongest components occur at frequencies between 600 and 900 cps. over a range of main shaft speeds between 88 and 115 r.p.m., and furthermore that the points corresponding to these components lie close to several of the sloping lines, in particular those corresponding to the 12th, 13th, 15th and 16th multiples of the h.-p. turbine speed and the 485th multiple of the main-shaft speed, there being 485 teeth on the main wheel. Vibration analysis showed the same general picture.

In order to ascertain with certainty whether the apparent integral multiples of h.-p. turbine speed were correct, and that the components did not arise from some unknown and completely different cause, a trial was made on vessel *B* with the results shown in Fig. 5. In this case a double-beam oscillograph was used in conjunction with a timing pick-up adapted to give a pulse once per revolution of the h.-p. pinion shaft in the manner described earlier in this paper. This confirmed that integral multiples of h.-p. turbine speed were present, the strongest being the 12th, with 11th and 13th sometimes nearly as strong. In fact only one component of any appreciable magnitude was *not* related to the speed of this turbine, namely a weak component of frequency 31 cps. observed at 119 r.p.m. (Multiple 240 of main shaft speed was also present, and identified by the fact that it produced a beat at about 1 cps. with the 7th multiple of h.-p. turbine speed.)

This indicated that the so-called noisy gearing was in fact entirely blameless and that the whole source of noise and vibration, and therefore probably the cause of the blade failures, was something connected with the h.-p. turbine and pinion line and associated with 11th, 12th and 13th multiples. This immediately fixed attention on the flexible coupling connecting the turbine shaft to its primary pinion, since this coupling was of the claw type having twelve teeth. No other source of 12th harmonic could be found and the presence of 11th and 13th harmonics could be accounted for by a slight skewing of either claw on its shaft as described earlier. It is therefore considered certain that the prime cause of noise and blade failures was this coupling since the fundamental natural frequency of the blades was such as to be excitable by one or other of these multiples at

almost any speed within the running range. There was evidence that the h.-p. turbine after pedestal was moving athwartships relative to the gear case by appreciable amounts, possibly due to steam pressure or temperature effects in a short horizontal receiver pipe between h.-p. and l.-p. turbines and provided with a flexible metallic bellows. On fitting additional stiffening to the pedestal the vibration was distinctly reduced and the noise reduction, although less, was appreciable.

There is, however, a probability that another fortuitous circumstance contributed, perhaps in large measure, to the severity of the conditions. This is the possible existence of a resonance in the turbine-coupling-pinion line taking place at about 800 cps. The noise intensities in vessel *A* arising from any of the various multiples tended to have a maximum value at about this frequency and in particular, the frequency corresponding to secondary tooth contact, while moderately strong at 87 and 99 r.p.m. was not found at 115 r.p.m. where the frequency would have been 920 cps. which is appreciably above the figure of 800 at which the noise was most intense.

It is not, however, so marked in vessel *B*, Fig. 5, in which the strongest component is at all times the 12th multiple of h.-p. turbine speed regardless of main-shaft revolutions. This high-speed shaft system has been critically examined for possible resonances and, of course, it possesses many resonant frequencies. No torsional resonance in the region of 800 cps. has been found by calculation and at the present time other forms of resonance are being sought for, in particular those corresponding to transverse bending of the shafts. This condition is very complex, since any transverse motion of the coupling immediately causes the teeth to ride on their radial edges (the teeth had no crowning on the driving flanks) and therefore applies additional twisting and bending moments to the shafts. There is, therefore, still the possibility of a vibration which is mainly transverse but is raised in frequency by the additional stiffness imparted to the shaft in bending by the torsional effects. If such a resonance exists, it would explain why the blade failures were confined to certain vessels of the class since the others had a coupling of slightly different design and proportions, although in principle it was the same and also had twelve teeth.

The disturbances produced by flexible couplings of the tooth type under conditions of misalignment may be greatly reduced by increasing the number of teeth in the coupling. New couplings are therefore being manufactured of the fine-tooth variety. This will have the double effect of reducing the magnitude of any disturbances that may arise from this source, and also raising their frequency so that it is above the natural resonant frequencies of all but the very shortest blades and these have usually higher damping in the blade fixings. It will not be possible to report on the success of these measurements until the first set of couplings has been made and fitted.

Example 2

Heavy vibration and noise were reported in a single-screw vessel. This machinery comprises h.-p. and l.-p. turbine driving through double-reduction gearing of the interleaved type. The trouble was also associated with cracked and broken teeth in the second reduction pinion on the l.-p. side. The noise was quite different from that of Example 1, being partly of the type associated with creep-cut gearing in which the noise spectrum is continuous without any specially prominent frequencies, and also accompanied by a heavy knock or rattle. The gearing was nearly twenty years old, being cut at a time when the accuracy of hobbing machines was far inferior to their present capabilities. The noise level was 112 db. on the l.-p. side and 109 db. on the h.-p. side at

full-power revolutions. The frequency of knock was too low to be detectable by the wave analyzer and no strong high-frequency components were found.

By lifting a light cover in way of the turning gear, which was situated on the h.-p. side of the gear case, the aft end of the intermediate gear shaft was visible and a chalk mark was made to assist observation of the rotation of this shaft. At full speed the rotation of the intermediate gear shaft was too rapid to be followed by the eye, but at speeds below about 40 r.p.m. it was noted that the knock occurred when the chalk mark was in the 8 o'clock position. An estimate of the time taken for sound emanating from the l.-p. side to reach this position indicated that the knock occurred when the mark was at 9 o'clock. When going astern the knock was also audible and occurred at the 10 o'clock position so that correcting for time delay, its true position was once again 9 o'clock.

This strongly suggested something of the nature of a bent intermediate gear shaft, and this was confirmed by an observation which could only be made once during the trial. When going astern during this voyage (which was not a special trial arranged for the observer's convenience) the shaft speed was usually low and only rose once to 42 r.p.m. On this occasion a definite whine was heard at some intensity. Its frequency was quickly measured with the small pitch pipe mentioned earlier and found to be approximately 800 cps. and immediately thereafter (about 15 seconds later) the speed of the intermediate gear shaft was measured by stop watch and found to be 2.56 revolutions per second. The frequency therefore corresponded to 313 impulses per revolution of the intermediate shaft, and was almost certainly the tooth-engagement frequency of primary gearing since the primary wheels had 317 teeth.

Now this whine is of no intrinsic interest, since it is very common for the tooth-engagement frequency to be audible at certain speeds due to a local resonance and the noise was not of any great intensity. However, the point which is of fundamental importance is that the whine was observed to be modulated with the frequency of intermediate-gear-shaft rotation, and varied in intensity and stridency once per revolution of this shaft. It was, therefore, a strong indication of an eccentric motion of the primary-wheel teeth, although of course it was not possible to say which of the two primary wheels was affected.

Examination of the primary pinion on the l.-p. side showed that the pinion teeth were very heavily ragged at the tips, this amounting to more than the ordinary rag, and giving the impression that a heavy "rolling over" effect had been taking place. A similar action was found on the associated primary-wheel teeth but in this case its intensity varied around the periphery of the wheel. When the primary gearing was subsequently set up in the hobbing machine for re-cutting, it was observed that the tooth annulus was definitely eccentric since with the hob just brushing at one part of the periphery, it was taking a heavy cut at the point diametrically opposite.

Example 3

During routine trials of a new vessel, the *Pametrada* representative noticed a peculiar rattling noise which was clearly audible in his room (situated about amidships on the starboard side) and also on deck in about the same locality. It was not sufficiently serious in intensity to indicate the likelihood of failures of any important part of the ship's structure, but was definitely disturbing in a vessel adapted for the accommodation of a small number of passengers. The frequency of the rattle was much too high to be counted against a stop watch and yet too low to be appreciated as a clear audible tone. Its principal characteristic was that it rose and fell

with a regular period of the order of eight seconds, being completely silent for two or three seconds before building up once more to maximum intensity. Timed over two successive intervals eight complete periods occurred in 59 and 58 seconds respectively. The Anson vibrometer was then applied to the deck planking but no indication was obtained. It was then applied to each of four pipes, connecting with numbers 4 and 5 double-bottom tanks and the cofferdams adjoining them, and on three of these no indication was observed, but on the fourth (vent from cofferdam between the two tanks) a frequency of 3700 vibrations per minute was detected with appreciable intensity (¼ in. double amplitude of reed).

At this time the main engines were stopped and only auxiliary machinery running. The frequency of rattle was therefore compared with that of the various auxiliaries and was found to be almost exactly twelve times the speed of the Diesel generators. According to their tachometer readings, the two which were in operation at the time were running at speeds of 308 and 309 r.p.m. respectively. The Anson vibrometer was now applied to the crankcase of one of these engines and the end of the reed was found to vibrate with a considerable amplitude which rose and fell between limits of ¾ in. and ¼ in. The period of the complete cycle was measured by stop watch, which indicated eight periods in 59 seconds, and 57 seconds at a second trial. The almost precise agreement with the period of beat observed in the rattle on deck is notable, although it should be emphasized that the rattle was naturally not audible in the engine-room due to the noise of the Diesel generators, and furthermore no trace of the slow beat was detectable in the engine-room except by means of the vibrometer. The frequency indicated by the vibrometer was slightly variable on different applications, the figures ranging between 940 and 960 vibrations per minute. It is almost certain that this corresponded to the firing frequency of the engines, which were of the three-cylinder two-stroke variety. The difficulty of obtaining a precise indication of frequency in the engine vibration is to be expected, since two engines were running at slightly different speeds, so that an instrument of high selectivity will give a beating response to any frequency in the range between the two values.

It was therefore diagnosed that the vent pipe from the cofferdam of numbers 4 and 5 double-bottom tanks was being excited by the firing frequency of the Diesel generators, and was vibrating either twice or four times this frequency according to whether the pipe was producing rattle by making contact with a loose clip on one or alternatively on either side. Time did not permit the tracing of this pipe to demonstrate with certainty whether the diagnosis was correct.

Example 4

Turbine-blade shrouding failures were reported in the l.-p. cylinder of a two-turbine installation driving a single screw through double-reduction gearing of the American nested type in which the two helices of primary wheels straddle the main-gear wheel. In this investigation, which has only recently been carried out, the wave correlator was used in conjunction with the wave analyzer to investigate both noise and vibration. Timing pulses were derived from both primary pinions, one intermediate-gear shaft and the main low-speed shaft. Even without the assistance of the wave analyzer, the correlator showed immediately that the greater part of the noise and vibration originated from the intermediate gear shaft. On detailed analysis, it was found that the principal components of noise and vibration corresponded to two multiples of the speed of this shaft. The first was the 50th multiple and the second was either multiple 125 or 126, though it

was not possible to ascertain with certainty which of the two was present. However, a weak component was definitely traced to the 75th multiple of the same shaft speed and another weak one was almost certainly the multiple 175, 176 or 177 of the same shaft speed. This suggests that the latter two components are in fact the sum and difference frequencies of the two principal components, and if so the multiple 126 is excluded. The multiple 50 is known to have been present in some gears cut on the same hobbing machine at a time when it was subject to certain errors which were later remedied, and was due to an error in one of the change wheels. This change wheel when used in an alternative position could have produced the 100th multiple and in fact more than a trace of an integral multiple, ascertained to be 100, 101 or 102. was also found during this trial but may have been merely the second harmonic of the 50th multiple. The multiple 125 was later ascertained to correspond to the worm-wheel error of the hobbing machine which was of the solid-table variety.

Measurement of the natural frequencies of turbine blades showed that the affected row had a first-order clamped-free frequency of 260 cps. of the major (peripheral)) type, and a first major clamped-pinned frequency of 1200 cps. (The ratio of these two is in close agreement with the value of $(2\frac{1}{2}/1.19)^2$ from Fig. 6, though exact agreement is not expected since the stiffening effect of the shrouding will probably differ slightly between the two modes). The frequency of 1200 cps., allowing for a small rise due to centrifugal forces when running, was equal to that of the strongest noise and vibration component (namely multiple 125 of primary-wheel speed) at shaft revolutions corresponding to 88 per cent. of full-power speed. Furthermore, the blade mode of vibration corresponded to motion in the peripheral direction and could therefore, readily be excited by torsional impulses transmitted from the gearing. The blade rows were shrouded and the shrouding itself was relatively highly stressed by centrifugal loading, and would not require a very large additional stress component to induce fatigue failure. It is therefore considered probable that the gearing errors were in fact the cause of the shrouding failures, although the noise and vibration intensities were not extremely severe, the noise level being about 104 db. at some distance from the gearing and 113 db. immediately on top of the gear case.

Example 5

A small water circulating pump in a central heating plant and driven by a fractional h.-p. motor gave rise to appreciable noise which, while not of high intensity in itself, was nevertheless distinctly uncomfortable in the comparative quiet of a drawing office. A very rough frequency analysis showed that numerous multiples of shaft speed were present, the 3rd being particularly strong but associated with 2nd, 4th, 5th. 6th and 7th, etc. It was realized that this harmonic series was compatible with the presence of 3rd, 6th and 9th. etc., multiples modulated by the first multiple. This strongly suggested the presence of a three-tooth coupling running out of alignment. On removing the coupling guard it was immediately clear that the alignment was not correct since adjacent coupling flanges differed in diameter by 1/16 in., and their rims were level *(a)* at the top, and *(b)* at the front, suggesting that the coupling flanges had been aligned on the assumption that they were of equal diameter. The coupling itself was of the floating-disc type and had three pins on each side. Such elaboration of method may seem unnecessary in such a minor component but in point of fact the diagnosis was completed within ten minutes and the misalignment proved in a few minutes more. The actual correction of alignment took a great deal longer since provision for adjustment was poor and not readily adaptable to accurate setting.

CONCLUSION

It is hoped that the examples quoted will serve to illustrate the manner in which the principles set out in the body of the paper may be applied to the diagnosis of vibration in marine-geared turbines. The importance of ascertaining whether or not any or all of the principal vibration components are associated with the various shaft periodicities cannot be over-emphasized. Even if the result is negative and the disturbance does not arise from rotating shafts, this very fact is extremely valuable evidence since it indicates that the source of disturbance must be sought elsewhere. After some years of experience on vibration phenomena, the Author can state with feeling that the identification, with certainty, of any single important connection between the vibration symptoms and the mechanical characteristics of the machinery can be of enormous help. Prior to development of the special equipment described in this paper, the principal characteristic of noise and vibration problems in running machinery has been their extreme complexity. The number of possible causes for any observed phenomenon is frequently so large that the observer is baffled and bemused at the very outset. The intense relief at the positive identification of even one prominent feature is difficult to describe, but will be appreciated by those who have themselves worked in this field. New instruments which are already in their infancy and will become still more powerful as their users become better accustomed to their capabilities, already give promise of achieving an entirely new standard of precision and certainty in vibration diagnosis, and, in the Author's opinion, will materially help in raising this field of study from something closely approximating to black magic to the higher level of scientific deduction.

ACKNOWLEDGMENTS

The author desires to thank the Research Director, Pametrada Research Station, for permission to present this paper, and his colleagues for very valuable assistance, in particular Mr. R. Catherall, B.Sc., to whose ingenuity and patience the development of the new electronic instruments owes much.

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Appendix I—Velocity as a Criterion of Vibration Stresses.
Appendix II—Natural Frequencies of Bars in Transverse Vibration.
Appendix III—Wave Analyzers.
Appendix IV—The Wave Correlator.

BOOK REVIEW

The RADIO AMATEUR'S HANDBOOK, 27th Edition, 1950; by the staff of the American Radio Relay League. Published by the American Radio Relay League, Inc., West Hartford, Connecticut. 736 pages, 1165 illustrations. Price $2.00 in the United States and its possessions.

REVIEWED BY: LIEUTENANT COMMANDER BURTON H. ANDREWS, U.S.N.

This new edition of the Radio Amateur's Handbook perpetuates a well-established and valuable institution. Long considered the Radio Amateur's bible, the Handbook has found the consistent favor not only of amateurs but also of nearly everyone interested in or concerned with practical electronics design, operation, or maintenance.

In keeping with its recognized progressive policies, the editorial staff of A.R.R.L. has carefully revised and brought up-to-date this latest edition. Reflecting the advances made in the field of electronics during the past year, much new material has been included. Particular emphasis has been given to the problems associated with the reduction of television interference. The recent adaptation of single-sideband telephony techniques to amateur use is brought out, and the characteristics of equipments suitable for such use are discussed. Receiver and transmitter designs incorporate new vacuum tube types and the best of modern circuitry. And again, reliability has been the keynote. The equipments described are tried and tested, and stability and ease of operation have not been sacrificed on the altar of novel but difficult and critical circuits.

The first four chapters present the elementary principles of electricity, vacuum tubes, circuit components, and radio communication. The material is presented concisely, in logical sequence, and with but little mathematics. Although by no means a complete textbook, these chapters make up an excellent primer covering in condensed form the important concepts necessary to an understanding of the basic electronic phenomena. Sufficient numerical examples, pictures, and schematic diagrams are included to ensure easy comprehension of the theories developed. Succeeding chapters supplement to a considerable degree the material contained in these first chapters.

The next eleven chapters deal with High Frequency (HF), Very High Frequency (VHF), and Ultra High Frequency (UHF) equipments. These chapters make up approximately two-thirds of the entire book, are very well written, easy to read, and reasonably extensive in scope. They cover receivers, transmitters, power supplies, telephony, and keying. The general pattern is followed of first expanding the principles given in the earlier chapters, analyzing their use in equipment design, and then giving complete design and constructional details for

specific equipments. Progressing from the very simple to the more complex designs, the range of material is such as to permit the novice to build and use excellent equipment while at the same time it provides a challenge to the abilities of the more advanced experimenter.

The succeeding eight chapters cover the assembly, test, and operation of amateur radio stations. Excellent data is offered for the design and construction of good test and measuring equipment, together with the basic techniques of electronics measurements. The construction and use of these equipments will go far toward bringing to the experimenter an understanding of good laboratory practices, as well as assuring him that his transmissions are within the legal limits of frequency, power, stability, and character of signal. These chapters also serve to bridge the gap between theory and practical application, in showing the integration of various equipments into a complete station and in giving the details of the on-the-air usage of such a station. Proper methods of message traffic handling by telephony and telegraphy are outlined. Members of the Amateur Radio fraternity have many times furnished the sole means of communication to and from disaster areas stricken by fire, wind, and water; considerable attention is given in these chapters to the construction and use of emergency portable equipments suitable for use on such occasions.

The final two chapters present miscellaneous valuable reference tables, data, and graphs, and a very comprehensive set of vacuum tube data charts. There follows an excellent and thorough index, well cross-referenced, and the last pages of the volume are devoted to advertising material supplied by the major manufacturers of reliable amateur and commercial equipments and components.

As in the past twenty-four years, this edition has preserved the virtue of a balanced combination of the theoretical and the practical. In spite of the very broad fields covered, sufficient material has been included to make the Radio Amateur's Handbook one of the most-used and most useful books in the library of anyone associated with electronics.

THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

29 West 39th Street, New York 18, New York

The Society of Naval Architects and Marine Engineers has announced that the Hydromechanics Subcommittee of its Technical and Research Committee has completed the first group of Model Resistance and Expanded Resistance Data Sheets prepared under a project on the compilation of data on ship resistance. This is a part of the continuous research being carried on under the auspices of the Society, and directed by outstanding members of the industry.

These new data sheets comprise pertinent information on 40 selected ship models, run at various establishments, and 40 sets of data obtained by expanding the test results of these models to 400-foot size.

The aim of this project is to make available to naval architects, marine engineers, and others interested in ship design problems, in a convenient and standardized form, comprehensive form data and actual test results on the multitude of ship models which have been and are being run in various American ship model testing establishments. This is the first time, so far as known, that such complete data,

carefully compiled and checked, covering the tests of such a large group of models, have been made available to members of the profession at large.

The models in the first or pilot group although limited in number in comparison to the thousands that have been tested, nevertheless represent a wide range in types and sizes of ship. This variety enabled the Hydromechanics Subcommittee to work out the method of presentation to better advantage, and to give it sufficient flexibility to cover all types that might be included in the future. As a result, the various models in the pilot group are unrelated to each other, but as the work of compilation continues series models and other related models can be included at any time.

The manner in which the model test data are presented on the sheets renders them available for any method of analysis other than the ones selected for the project as representative and generally useful. The expansion of the model data to 400-foot ship scale by several methods affords a wide basis for comparison.

Standard forms for the presentation of the model resistance data and the resulting expanded resistance data have been developed.

The data sheets are accompanied by copious Explanatory Notes, giving a full description of the various entries, a statement of the methods and procedure employed in working up the data, a complete list and definition of all symbols, and all other information necessary to enable immediate use of the data sheets.

Model Resistance and Expanded Resistance Data Sheets, with Explanatory Notes, are now available for sale to all individuals or organizations interested in this phase of the ship design problem. Inquiries or orders should be addressed to the office of the Society, at 29 West 39 Street, New York 18, New York. Cost to members of SNAME is $4.00, to non-members $5.00.

ERRATA

In Part Two of the issue of November 1949, the reprinted article "Surface Treatment of Diesel Engine Parts" was credited to the October 1949 issue of MOTORSHIP. The correct reference should have been the August 1949 issue of MOTORSHIP.

MEMBERSHIP CHANGES

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Gang Foreman, Diesel Shop, Pennsylvania R.R.
Mail: 2529 So. Second St., Steelton, Pa.

Brown, James Andrew, Commander, U.S.N.,
1013 N. Edgewood St., Arlington, Va.

Buck, R. Admiral, Walter A., U.S.N., Ret.,
Operating Vice President, RCA Victor Div.,
Camden, N. J.

Burroughs, Robert E. A., Commander, U.S.N.,
Manager of Engineering, Aircraft Gas Turbine Division
General Electric Co., 920 Western Ave., W. Lynn, Mass.

Cassidy, Marcellus William, Commander, U.S.N.R.,
Industrial Manager's Office, Room 226, Bldg. 1—
U. S. N. T. Center, Great Lakes, Ill.

Clark, Edward Coe, Lieutenant Commander, U.S.N.R.,
Crane Co., 1225 Eye St., N. W., Washington, D. C.

Coil, Tom Paul, Commander, U.S.N.,
5811 Namakagan Road, Washington 16, D. C.

Cook, Melvin S., Ensign, U.S.N.,
U.S.S. *Fechteler* (DD 870), c/o Fleet P. O., San Francisco, Calif.

Dahl, Carl B., Lieutenant Commander, U.S.N.R.,
Design Calculator, Fairbanks, Morse & Co.,
Mail: 1105 Chapin St., Beloit, Wis.

DeSoboll, F. Stephen, Lieutenant Commander, U.S.C.G.,
P. O. Box 2515, Station B., San Francisco 26, Calif.

Dorsey, Lawrens, Ensign, U.S.N.,
U.S.S. *Kearsarge* (CV 33),
Puget Sound Naval Shipyard,
Bremerton, Wash.

Dunstan, Whitnet Atherton, Ensign, U.S.N.,
Matson Navigation Co.,
Mail: 206 Ave. G., Rodondo Beach, Calif.

Evans, John Harry, Asst. Professor, M.I.T.,
Mail: 31 Totnes Road,
Braintree 84, Mass.

Fitzsimmons, Andrew M. R., Capt., U.S.N.,
Office of Industrial Survey, Room 1103
Navy Dept., Washington, D. C.

Frobe, Raymond J., Lieutenant Commander, U.S.N.R.,
Arkansas Power and Light Co.,
Lake Catherine Station, Hot Springs, Ark.
Mail: 707 S. Main St., Malvern, Ark.

Gomez, Rodriga Janrique, Ensign, U.S.N.R.,
1030 E. Lee St.,
Tucson, Arizona

Grankull, Elmer Arthur, Lieutenant, U.S.N.R.,
Electrical Engineer, Puget Sound Naval Shipyard
Mail: 308 S. Charleston Ave., Bremerton, Wash.

Hamm, William Francis, Ensign, U.S.N.R.,
Marine Engineer, 58 Essex St., Marlboro, Mass.

Huske, John Caldwell, Capt., U.S.N.,
6207 N. 19th St.,
Arlington, Va.

Kelly, Robert Lester, Ensign, U.S.N.R.,
Naval Architect,
147-B, Oak St., Bremerton, Wash.

Ladd, Edward Fedde,
Reqn. Engr. Marine Systems Div.
Aeronautics & Ordnance Systems Divs.,
General Electric Co.,
Mail: 1041 Lexington Ave., Schenectady 9, N. Y.

Lowers, Horace Robinson, Lieutenant, j.g., U.S.N.R.,
Mech. Engr. Bureau of Aeronautics, Navy Dept.,
Mail: 73 Forrester St., S. W., Washington, D. C.

Madden, Robert Bowman, Commander, U.S.N.,
Code 140, Puget Sound Naval Shipyard,
Bremerton, Wash.

Mele, John Joseph, Lieutenant, j.g., U.S.N.R.,
General Maintenance Foreman, Ramey Air Force Base
Mail: P. O. Box 15, APO 845,
c/o Postmaster, New York, N.Y.

Mendel, Frank Stanley, Ensign, U.S.N.R.,
Asst. Electrical Engr., Allis Chalmers Mfg. Co.
Mail: 1579 So. 80th St., Milwaukee 14, Wis.

Molloy, George Pryor, Lieutenant, j.g., U.S.N.R.,
In charge Field Eng. on Electrical Glass,
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Mail: 90 Bell St., Belleville, N. J.

Noe, Abram W., Ch. Mach., U.S.N.R.,
in charge Rep. & Air Comd. Shop,
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Mail: 6611 Bullock St., San Diego 11, Calif.

Rivers, William Jacocks, Lieutenant Commander, U.S.N.,
Box 126, N. O. B., Ind., Navy 626
c/o Fleet P. O., San Francisco, Calif.

Schumacher, Gilbert F., Commander, U.S.C.G.,
Head Repair & Material Sec., U.S.C.G.
Mail: 5514 Small Wood Drive,
Green Acres, Md., 16 D. C.

Wheeler, Frank Knowles Blasdell, Commander, U.S.N.,
CNO (Op 413) Navy Dept., Washington, D. C.

York, William Eugene, Ensign, U.S.N.R.,
Marine Engr. Union Oil Co. of Calif.
Mail: 54-23 The Toledo, Long Beach 3, Calif.

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Adair, James M., Vice President Sales & Engineering
Star Electric Motor Co., 200 Bloomfield Ave.,
Bloomfield, N. J.

Allen, Robert Chester, Manager and Chf. Engr.
Turbo-Power Div. Dept., Allis Chalmers Mfg. Co.
Mail: 649 North 68th St., Wauwatsa 13, Wis.

Bigger, Michael M., Mech. Engr., Naval Radiological
Defense Laboratory, San Francisco Naval Ship Yard
Mail: 21 Highland Ave., Apt. 1, Burlingame, Calif.

Bielefeld, Richard H., Asst. Engr. Electrical Contracts,
Allis Chalmers Mfg. Co.
Mail: 2714 West Melvins St., Milwaukee 11, Wis.

Brace, Norman G., Bureau of Ships, Navy Dept.
Mail: 719 Richmond Ave., Silver Spring, Md.

Burkhardt, Donald Taylor, Asst. to Machinery Supt.
Bath Iron Works Corp.
Mail: 1299 High St., Bath, Me.

Camp, Philip Littlehale, Supervisor of Application
& Requisition Engrg., Air Circuit Burner Div.
General Electric Co.
Mail: 6901 Elmwood Ave., Philadelphia, Pa.

Collins, Laurence Joseph, Div. Engr.,
Gear Engineering Div. General Electric Co.,
920 Western Ave., Bldg. 5-64-G,
West Lynn, Mass.

Dahlstrand, Josef Y., Vice President,
Universal Gear Corp., 145 E. 19th St.
Indianapolis, Ind.

Day, George Earlon, Vice President in Charge of
Engineering Sig-Trans, Inc., Amesbury, Mass.
Mail: 6½ S. Hampton Road, Amesbury, Mass.

Fraser, Edwin James, Technical Supervisor, Piping Navy Dept.
Mail: R. D. #3, Kimmell Apts., Mechanicsburg, Pa.

Godfrey, John W., Asst. Chief, Product Engineer,
Industrial Section, Harrison Radiator Div., GMC
Lockport, N. Y.

Grassman, Herbert Stuyvesant, Advisory Engr.,
Taylor Forge & Pipe Works,
Mail: P. O. Box 485, Chicago 90, Ill.

Gross, Frits A., Dept. Manager
Radar Engineering Dept. Raytheon Mfg. Co.,
148 California St., Newton, Mass.

Hopkins, Gordon C., Sales Manager
RCA Victor Div., Room 502, 1625 K St., N. W.,
Washington, D. C.

Hudnal, James B., Engr. for Ship Electronics.
Bureau of Ships, Navy Dept.
Mail: Apt. 1, 1528 17th St., N., Arlington, Va.

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Federal & Marine Engine Div., General Electric Co.
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101 N. Charles St., Baltimore, Md.

Woodard, Henry William, Sonar Project Engineer
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RCA Victor Div., Bldg. 13-7-RCA Victor Div., GMC
Camden, N. J.

Wright, Justice Wesley, Mgr. Diesel Engine Div.
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Tyndale-Biscoe, Alec Julian, Capt. (E) R.N.,
British Joint Services Mission (Navy Staff)
1910 K St., N. W., Washington, D. C.

Voges, Henry E., Executive Representative
Walworth Co., 810 18th St., N. W.
Washington, D. C.

Woods, John, Jr., Sales Engineer,
General Electric Co., 806 15th St., N. W.,
Washington, D. C.

TRANSFERRED ASSOCIATE TO CIVIL

Powell, Shannon Curtis, Professor, Dept. of N. A. & M. E.
Massachusetts Institute of Technology
Cambridge 39, Mass.

DEATH NOTICE

The Society has learned with regret of the death of

MR. HARRY BROWN

Chairman of the Board of
Foster Wheeler Corporation

Civil Member

ASSOCIATION NOTES

Original Articles

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

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4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

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Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

Subject Matter and Authors

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(*a*) What subject would you, as a member, like to see covered by an article in the JOURNAL.

and

(*b*) Who do you consider best qualified to prepare such an article authoritatively, and what is his address. (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

Photographs

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

Notice of Deaths

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

Society Lapel Button

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.

It is available to all members at fifty cents (50c) each.

LIFE MEMBERSHIPS

Any member may now purchase a life membership under the following rules:

(*a*) Any dues paid for the year in which life membership is purchased will be credited to the cost of the life membership.

(*b*) Life memberships are non-transferable and terminate with the death of the member.

(*c*) No refund will be made on account of death or resignation of a life member.

(*d*) Life membership will vary in cost with the age of the member at his next birthday following application, as follows:

Over 55 years	$125.00
40 to 55 years	175.00
25 to 39 years	225.00
Under 25 years	250.00

(*e*) All life memberships paid in shall be protected by a transfer on the books of the Society of Series G, U. S. Government 2½ per cent bonds. Thus a life membership fund will be established. It will be credited with moneys paid in for life memberships and debited with the regular amount of annual dues for each life member each year.

LOCAL CHAPTERS

Local Chapters of the American Society of Naval Engineers may now be authorized by the Council. The following rules for recognition of Local Chapters have been adopted by the Council:

(*a*) Twenty or more regular members, Naval or Civilian, may apply for a Charter as a Local Chapter.

(*b*) Each application must be accompanied by a list of members and a copy of local By-laws. The latter must meet the following minimum requirements.

1. All financial dealings of the local Chapter must be independent of the Society and can in no way obligate the Society as a whole. The local chapter shall be set up on a non-profit basis. All accounting shall be local.

2. Each active local Chapter shall receive from the Society a grant of $0.50 per year for each naval member, civil member or associate who resides in the local area and who associates himself with the local Chapter. This grant may be used by the local Chapter to defray any necessary expenses, including (a) Letterheads, (b) Notice cards, (c) Postage, (d) Minimum rental allowance, (e) Secretarial service (part time).

3. Any technical papers presented at local chapter meetings shall be submitted to the Secretary-Treasurer of the Society for consideration for publication in the JOURNAL. Any paper accepted will be paid for at regular rates.

4. Each local chapter shall submit a quarterly report of its activities to the Secretary-Treasurer. These reports must reach Washington, D. C. prior to 15 January, 15 April, 15 July and 15 October. This report shall contain a list of active local members as of the first day of the quarter for the purpose of computing the amount of grant.

5. Associate members of the Society shall be eligible for membership in a Local Chapter, entitled to all the privileges of other members except voting and holding office.

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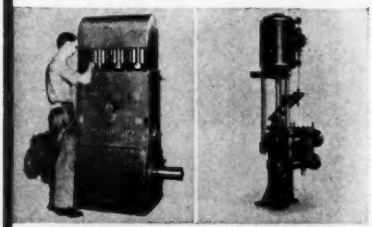

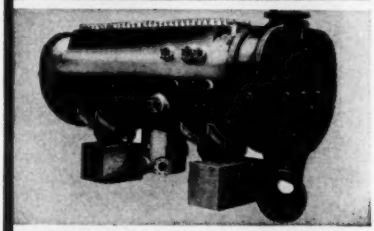

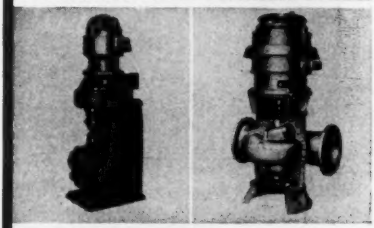

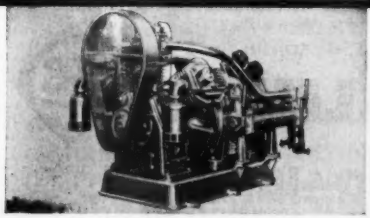

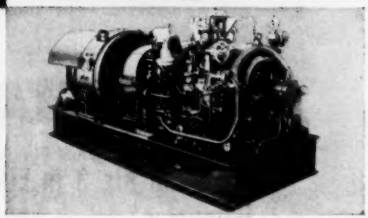

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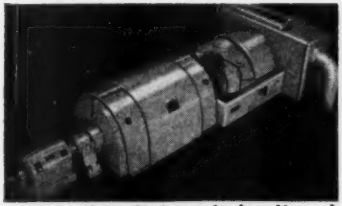

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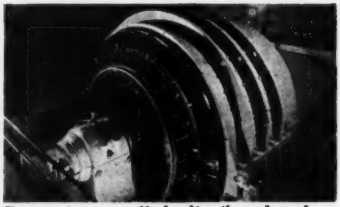

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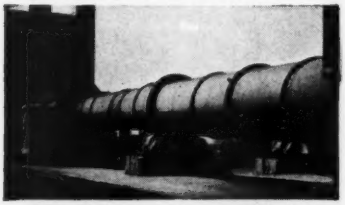

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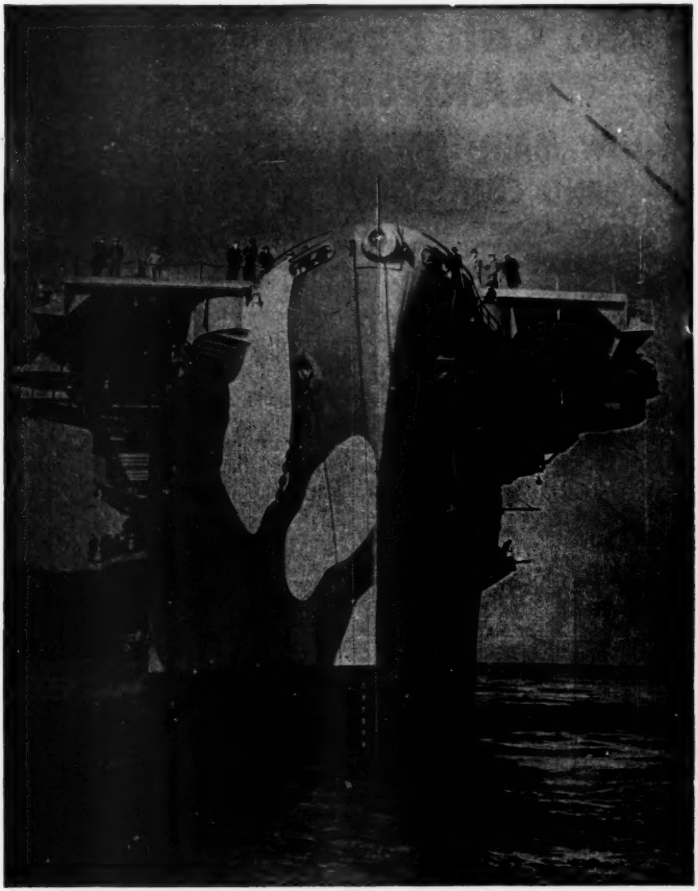

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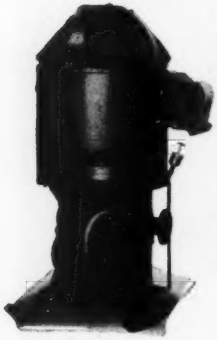

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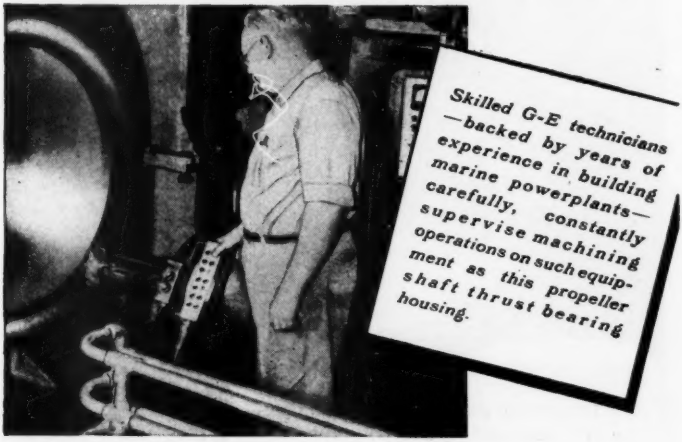
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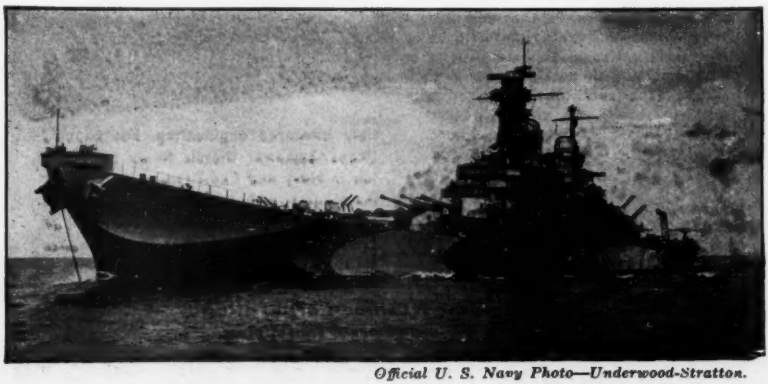

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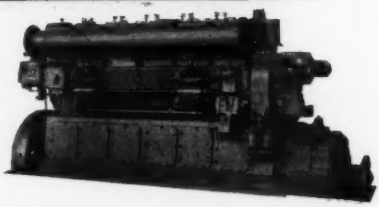

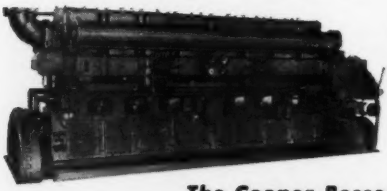

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Persons in civil life whose knowledge of engineering is such that they can cooperate with Naval engineers in the promotion of professional knowledge may be eligible as civil members. They shall have been in the active practice of an engineering profession for at least eight years and in responsible charge of important work for five years, and shall be qualified to design as well as to direct engineering work. Fulfilling the duties of a professor of engineering who is in charge of a department in a college or school of accepted standing shall be taken as an equivalent to an equal number of years of active practice. Graduation from a school of engineering of recognized standing shall be considered as equivalent to two years of active practice. Persons eligible as civil members may be admitted upon application and payment of annual dues, provided that the application is accompanied by the recommendation of two members and provided that the application shall receive the approval of a majority of the Council.

Persons in civil life who are not eligible for civil membership, but who are especially interested in naval matters or the merchant marine may be eligible as associate members. Commissioned officers of the United States Army and of foreign military and naval services may be eligible as associate members. Persons eligible to associate membership may be admitted upon application and payment of annual dues, provided the application have the recommendation of a member and provided the application shall receive the approval of a majority of the Council, except that in the case of commissioned officers of the United States Army and of foreign naval and military services, the recommendation of a member will not be required.

Associate members shall be entitled to all the privileges of other members except voting and holding office.

The annual dues shall be $7.50 payable on 1 January in advance.

APPLICATION FOR MEMBERSHIP IN THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

Date _____

I hereby make application for membership* in the American Society of Naval Engineers and submit the following information:

For Naval Membership

(First) (Middle) (Last)

Name _____

Rank _____ File No. _____

Business connection and position, if any _____

For Civil Membership

(First) (Middle) (Last)

Name _____

Years in engineering work _____

Years in responsible charge of important work _____

Present business connection and position _____

Recommended by (two members) _____

_____

For Associate Membership

(First) (Middle) (Last)

Name _____

Rank, if Commissioned Officer of U. S. Army or of foreign military or Naval service _____

Business connection and position _____

Recommended by (one member) _____

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Signature of Applicant

Address for Journal and Mail

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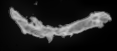
MAIL TO SECRETARY-TREASURER
THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.
BUREAU OF SHIPS, NAVY DEPT., WASHINGTON, D. C.

* See reverse side for required qualifications for various classes of membership



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